

CRITICAL FLOW OF WET STEAM THROUGH A
WELL ROUNDED ORIFICE

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1971

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WELL ROUNDED ORIFICE

by

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A thesis submitted in partial fulfillment
of the requirements for the degree of

MASTER OF SCIENCE

IN

MECHANICAL ENGINEERING

UNIVERSITY OF WASHINGTON

1971

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Abstract

The experiment explores the critical flow properties of two phase steam in a well rounded converging orifice. Mass flow rates were obtained over a quality range of 0.34 to slight superheat at pressures of 20 to 60 psia. Attempts were made to correlate previously determined velocities of wet steam with mass flow rates in order to evaluate the liquid-vapor velocity ratio, or slip ratio. It became apparent, however, that limitations imposed by the orifice size yielded nonuniform values of the slip ratio across the orifice.

Acknowledgment

My deepest appreciation is extended to Professor J. C. Firey for his encouragement and aid on this experiment.

Many thanks are due to W. Piispanen and W. Schoenfeld for construction and maintenance of the apparatus.

I also appreciate the time taken by Professors P. J. Waibler and R. W. Moulton to review this manuscript.

Equipment and supplies were made available through the Mechanical Engineering Department of the University of Washington. My own presence at the University of Washington was made possible through the Navy under the Navy Enlisted Science Education Program. None of the contents of this thesis are to be construed as naval policy.

Nomenclature

A	=	area, ft^2
C_p	=	specific heat, $\text{BTU/lbm}^\circ\text{F}$
G	=	mass flow rate per unit area, $\text{lbm/ft}^2\text{sec}$
h	=	specific enthalpy, BTU/lbm
K	=	slip ratio
\dot{m}	=	mass flow rate, lbm/sec
P	=	pressure
T	=	temperature, $^\circ\text{F}$
V	=	velocity, ft/sec
v	=	specific volume, ft^3/lbm
X	=	quality
Z	=	compressibility factor
ρ	=	density, lbm/ft^3

Subscripts

B	=	back
c	=	critical
f	=	liquid
g	=	vapor
i	=	inlet
o	=	outlet
S	=	supply
T	=	test
t	=	total
w	=	water

Introduction

The purpose of this thesis is to determine experimentally the critical flow characteristics of very wet steam through a well-rounded converging orifice.

Critical flow through an orifice is achieved by lowering the mass pressure to point at which no further increase in mass flow rate occurs with constant upstream pressure. According to Reynolds (1), the fluid velocity at critical flow is equal to the acoustic velocity in that medium. Acoustic velocity is the velocity at which a small pressure wave moves through the medium. Wet steam is defined as vapor with suspended liquid droplets.

Experimental and theoretical work continues on two phase steam flow as well as other fluids and gas-solid mixtures. The bibliography supplied with Reference 2 is a recent indication of some of the work being done with one-component, two-phase critical flow in nozzles and orifices. The principle references used in this study are W. G. England (3), O. E. Trapp (4), and S. G. Steltz (5). England and Trapp made experimental determinations of the acoustic velocity in wet steam. The acoustic velocity was found by measuring the time a single pressure wave took to pass two transducers. Their conclusion was that the measured sound wave was the wave that passed through the vapor only. The quality of steam investigated range from 0.20 to a slight superheat.

Steltz carried out a computer study to determine the acoustic velocity in wet and superheated steam undergoing isentropic expansion. His results showed a discontinuity in the value of the acoustic velocity of steam at the saturation line. England studied this saturated to superheated transition region by experimental means and did not find this

discontinuity. Steltz's work also showed a substantial decrease in the acoustic velocity of wet steam with decreasing quality. Trapp's experimental results in very wet steam do not confirm these results, but rather showed the acoustic velocity to be almost constant at a constant test pressure, decreasing only slightly with decreasing quality.

The present experiment determines the mass flow rate of wet steam through a well rounded orifice in critical flow at low pressures (20 to 60 psia) and low qualities (0.34 to a slight superheat). Pressure ratios across the test orifice ranged through the critical flow region into the noncritical flow region. Wet steam was made by expanding steam through a Mach 1.2 nozzle and mixing the steam with water from a high pressure spray nozzle, thereby suspending droplets of liquid in the vapor. Some of this steam water mixture was then passed through the test orifice. The mass flow rate was measured and compared to the pressure ratio across the orifice in order to determine the critical mass flow rate and the critical pressure ratio.

The determination of the actual fluid expansion process through the orifice is of paramount importance, since the fluid velocities, specific volumes, and the quality in the throat must be known accurately to predict the critical mass flow rate.

The equilibrium process for the expansion of steam for the critical flow condition is shown in the Mollier chart of Fig. 1. Under equilibrium, isentropic expansion the steam is seen to decrease in quality as it passes through the orifice. The enthalpy decreases while the kinetic energy increases as the fluid accelerates. After the orifice, nonisentropic expansion continues with recovery of enthalpy as the fluid decelerates,

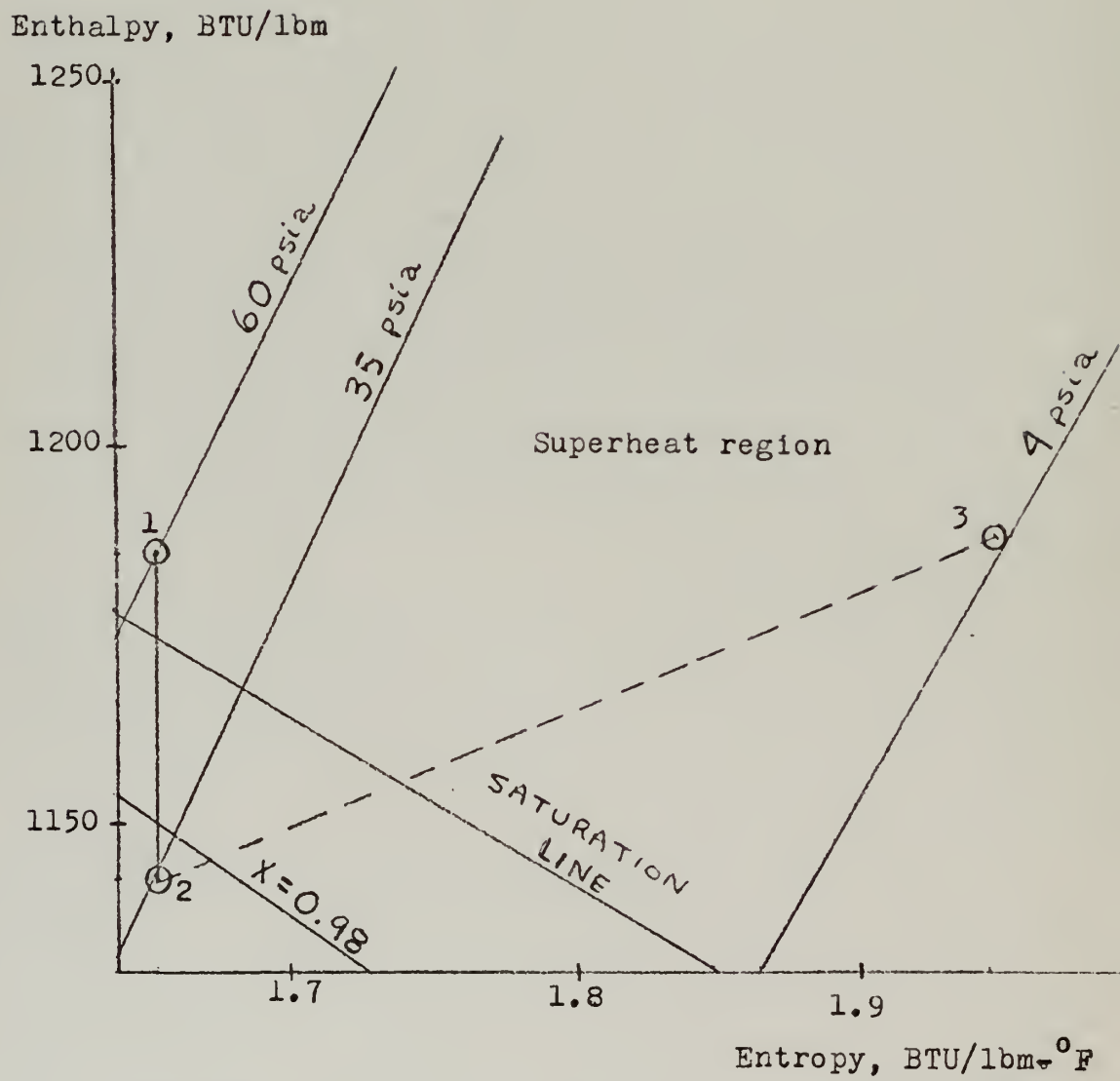


Fig. 1 Mollier Process Diagram

thus the entire process is ideally isoenthalpic.

Other possible processes exist for expansion within the orifice, all of which involve nonisentropic processes. The assumptions concluded to be representative of the process in this experiment consist of the following:

1. The average velocities of the two phases are not equal.
2. Phase change in the throat of the orifice is substantially less than that required by isentropic equilibrium.
3. The expansion process is nonisentropic, due in large part to the friction between the vapor and liquid phases.
4. The vapor properties are allowed to approach those of a perfect gas during expansion.
5. There is no heat transfer.

If the liquid and vapor velocities were to be calculated by taking into account the acceleration of the two separate phases due to a pressure gradient, the ratio of liquid to vapor velocity (slip ratio) would be approximately 0.001. Results calculated from experimental data show the slip ratio to be approximately 0.02, indicating that there is some aerodynamic drag on the suspended liquid droplets by the vapor as the whole stream moves through the test orifice.

An isentropic expansion process in the orifice demands that there be some condensation of the vapor phase in the throat, but there is evidence that no condensation occurs within the orifice. The Wilson Line (6) indicates that condensation does not occur until an enthalpy drop of some 60 BTU/lbm still be somewhere below point 2 in the test case (Fig. 3) since $h_1 - h_2 = 45$ BTU/lbm. The previous statement assumes the saturated state to be at an enthalpy equal to or lower than test section enthalpy. It is

also necessary to assume that droplets in the wet steam do not provide a nucleating surface for condensation in the orifice. Transit time through the orifice is on the order of 4×10^{-5} seconds which compares to the time interval of 1×10^{-5} for a shock compression wave (6). Equilibrium conditions cannot be used to correctly calculate conditions in the shock wave because of this short duration, so it follows that equilibrium conditions are probably not found in the orifice.

Some difficulty is met when trying to match mass flow rates to the available area using acoustic velocities determined by Trapp and England. One solution which yields plausible results takes into account the compressibility factor, Z . During the expansion of the vapor through the orifice assume that the value of Z approaches unity, in other words, that the vapor has the properties of an ideal gas rather than steam table properties when suddenly expanded. If the specific volume of the vapor is reduced by the value of the compressibility factor, then the area occupied by the vapor portion is reduced to a value approximately 5% below the value of the available area, in effect giving an orifice coefficient of 0.95. This value of the orifice coefficient conforms to that given in ASME Fluid Meters for rounded orifices. (7)

The calculations from the data of this experiment for the average velocity under critical flow conditions are made under the assumption that both the liquid and the vapor phases are traveling at the same velocity in the test orifice. This is undoubtedly not true, so the critical velocities presented in the chapter on results are only for purposes of comparison to acoustic velocities.

Apparatus

A schematic diagram of the apparatus is shown in Fig. 2. Steam was drawn from the distribution system in the engineering laboratory and mixed with water from the city water supply to make the steam-water mixture. City water was also used for condensing the steam.

Mixing water supply

Water from the city lines flowed into a pump driven by a 2 HP motor. The output pressure of the pump was controlled by a recirculating bypass line and limited by a relief valve. The water passed through a three-stage filter and on to a heat exchanger which raised the water temperature to approximately 195°F. High water temperatures are needed to minimize steam condensation on the water droplets. Finer water droplets allowed attainment of lower qualities in the test section. The water was then piped to the water spray nozzle inside the six inch cast iron pipe "T." The water spray nozzle was one of several available oil burner nozzles. Sizes used ranged from 17.5 GPH to 100 GPH with a spray pattern that was essentially a hollow cone of 80° included angle. Mounted in front of the nozzle was a 40° cone (Fig. 3) to deflect the steam away from the center of the water nozzle. The deflected steam met the water spray at an approximate 90° angle.

Steam supply

Dry saturated steam came from the University of Washington Power Plant to the Mechanical Engineering Laboratory distribution system at 180 psig. A regulator valve reduced the pressure to the desired range of 79 to 165

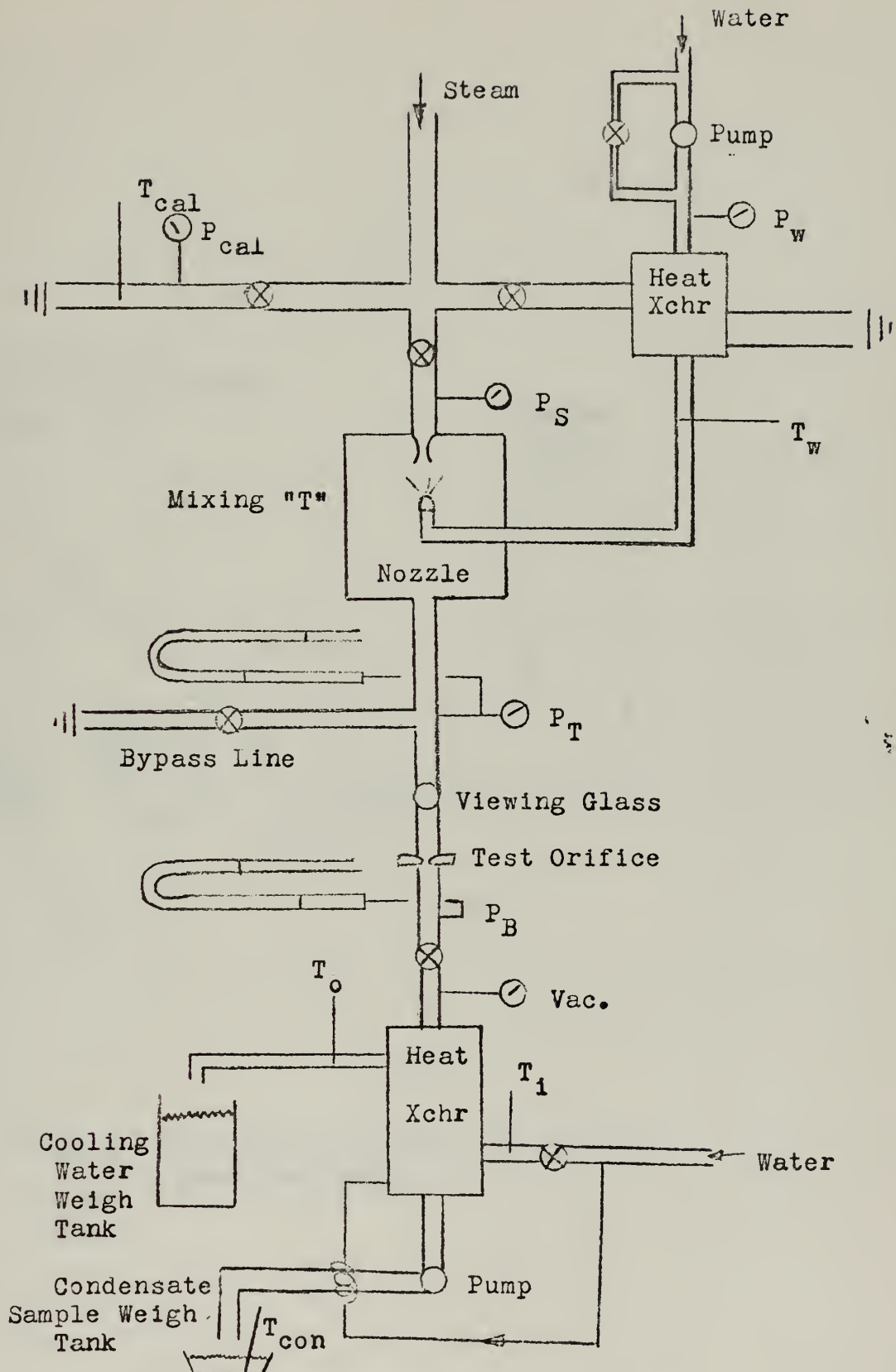


Fig. 2 Schematic Diagram

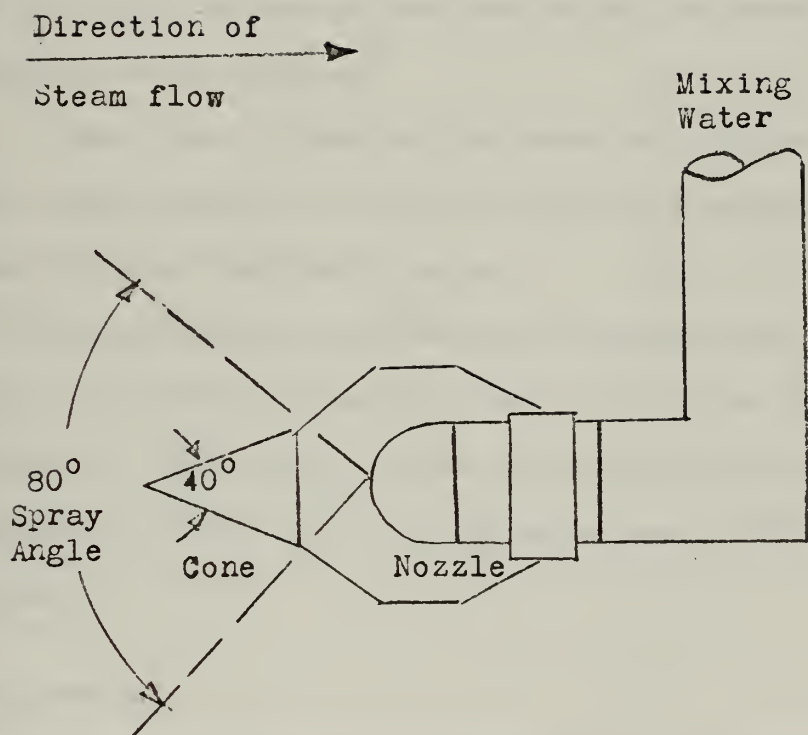


Fig. 3 Mixing water spray nozzle with steam deflector

psig. Steam traps, devices that remove condensate in steam lines in order to maintain dry saturated steam, are built into the distribution system. By closing off the steam traps, steam of less than 0.96 quality was attained at the test apparatus. Supply steam quality was measured by a throttling calorimeter sampling from the supply lines. The exhaust from the throttling calorimeter was used to heat the mixing water in the heat exchanger described above.

A drain valve, placed at the lowest point in the supply system, was left slightly open at all times to prevent an excessive buildup of condensate in the distribution system.

The steam supply line then led, through a normally-open isolation valve, to converging-diverging steam nozzle in the "T." Upon passing through the steam nozzle, the accelerated steam impinged on the cone described above and mixed with the water spray, forming the steam-water mixture.

Test Section

The test section extends from the "T" to the test orifice flange. Included is a safety valve and a two inch pipe cross with a pressure gage for monitoring the test section pressure and a bypass line with a globe valve for test section pressure control. The bypass line was used to drain any water condensing on the walls of the test section before the test orifice and as the primary means of controlling test section pressure. Just downstream from this cross was another horizontally mounted pipe cross which had glass mounted in the unused openings. This allowed a visual check of the test section to insure that slugs of water were not building

up in front of the test orifice. By adjusting the steam supply pressure and the bypass valve a minimum of water was left in the test section. This water appeared in a small pool in the bottom of the cross. The surface of the water was smooth and apparently stagnant except in the case of the wettest steam conditions. With the aid of this viewing glass, the nature of the steam-water mixture could be seen. The mixture appeared as a heavy mist, the water droplets about the size of misty rain.

A brass plate 0.300 inches thick with a rounded-inlet converging orifice completed the test section. The orifice throat had a diameter of 0.300 inches. The radius of curvature of the inlet was also 0.300 inches. The brass plate was secured between two pipe flanges.

Exhaust and condensing section

After expansion through the test orifice, the steam entered another two inch pipe which led to a fixed-stem globe valve. The back pressure on the orifice was controlled by regulating this valve. Just upstream of this point, a mercury manometer was tapped in to measure back pressure.

A vertically mounted shell and tube heat exchanger was down stream of the regulating valve. City water was supplied to the shell side of the heat exchanger. Because the heat exchanger proved unable to condense all of the steam, further cooling of the steam took place in the sampling tank to be described below. The inlet and outlet temperatures of the cooling water were measured with precision calorimeter thermometers graduated to 0.1°F . The mass flow rate of water was measured on a 400 pound capacity scale, graduated in 0.25 pound divisions.

A wetvacuum condensate pump delivered the condensed steam to a large sampling tank. The large volume of the sampling tank provided recirculating water to keep the condensate pump wet and cool, as there was insufficient condensate to keep the pump wet. A standpipe in the tank maintained a constant level, with the excess, which was equal to the condensed steam, drained off to a weighing point.

As mentioned above, the heat exchanger was incapable of condensing all the steam. To correct this, cooling coils were immersed in the sampling tank. The coils cooled the recirculating pump water which condensed the remaining steam by direct contact. In order to measure the energy thus removed, the water sent through the coils was piped to the inlet end of the heat exchanger where it mixed with the rest of the cooling water, thus contributing to the increase in temperature and total mass flow of the cooling water. The temperature of the sampling tank was taken to be the temperature of the condensed steam.

Pressure and temperature gages

There are three pressure regions in the apparatus. These are:

a) supply pressure, b) test section pressure, and c) exhaust section pressure.

The steam supply pressure was measured by a Bourdon gage. In order to determine the state of the steam, a throttling calorimeter was provided.

A Bourdon gage with half pound gradations was used in the test section. The accuracy of this gage was checked twice during the test period on a dead weight tester. Also provided was a connection to a 50 inch mercury manometer whose other leg was open to atmosphere. This manometer was used on the 29.3 psia and 20 psia tests to serve as a check on the Bourdon gage.

Since steam would condense on the mercury, thereby changing the effective height of the mercury column, a known head of water was put on top of the mercury column on the steam side of this manometer.

The exhaust section pressure was monitored on another 50 inch manometer. A known head of water was put over both legs of this manometer. Downstream of the back pressure control valve a Bourdon vacuum gage was placed. This gage served little more than to indicate that the condensate pump was working. The final condensate temperature was measured on a standard laboratory thermometer in the sampling tank.

The water pressure on the spray nozzle was measured by a Bourdon gage mounted downstream of the filters. The temperature was measured by a bimetallic thermometer just after the heat exchanger.

The cooling water temperature in the heat exchanger was monitored on two precision calorimeter thermometers graduated in $1/10^{\circ}\text{F}$ divisions. The quantity of cooling water was determined on a 400 pound scale with $1/4$ pound divisions. The steam sample weight was determined on a 10 pound scale with $1/1000$ pound divisions.

Test Procedure

The test procedure was devised to cover the complete range of testing from superheat to low quality at each pressure. The procedure also provided continuous determination of test section quality by calculating the heat energy transferred from the steam to the cooling water in the heat exchanger and the sampling tank.

For superheated steam tests, that is, with no water spray, steam was expanded into the test section and again into the exhaust section. For this procedure, all steam traps in the distribution system were left open to maintain dry supply steam. Steam supply pressures were set equal to the pressure required to successfully run tests with the water spray on and maintain the desired pressure in the test section. The bypass valve was used to provide exact control of the test pressure. A warmup period of at least one hour was allowed. Initially the back pressure regulating valve was wide open in order to obtain the lowest back pressure. After recording the necessary pressures and temperatures, the steam condensate was collected over a specified period of time and weighed. When the sample had been collected the back pressure regulating valve was partially closed. The amount of closure was controlled by observing the back pressure manometer. The procedure was then repeated for several points at increasing back pressure until a pressure ratio (back pressure divided by test pressure) of about 0.65 was reached. For several tests, the procedure was reversed to check values with decreasing back pressure.

For test runs with low quality steam the same procedure was followed, but the steam traps were closed in order to obtain wet supply steam. Warm-up was extended to about one half hour after the throttling calorimeter had

ceased to register any change. The throttling calorimeter could not operate properly with wet supply steam, so supply steam quality could not be calculated. The quality was, however, below 0.96. During the warmup period the water spray was turned on. The water spray pressure was set at approximately 120 psi above test section pressure. High water temperatures and high water pressures provided good mixing.

As in the dry steam runs, the tests were begun with the back pressure regulating valve wide open and subsequently closed in stages to obtain higher back pressures. At this point, the unexpected occurred. As the back pressure was increased, the test section pressure also increased, although very slightly, necessitating a wider opening of the bypass valve to maintain a constant test section pressure. The data shows that the quality was not affected by using a wider valve opening, which strengthens our conviction that we were able to avoid slugs of liquid passing through the orifice. The results obtained by leaving the bypass valve at one setting while changing the back pressure were measured and are presented in Fig. 10.

Results

The mass flow rates compared to pressure ratio, test section pressure, and test section quality are shown in Figures 4-9. These graphs may be used to predict flow rates through small rounded orifices at low pressures, but we place no confidence in extrapolation of the graphs beyond this region because of boundary layer effects in small orifices and the indeterminate slip ratio. To conclude, some remarks are presented concerning the value of the critical velocity.

Figures 4-7 show the measured change in mass flow rate with change in orifice pressure ratio at five different qualities for each of four different test section pressures. Figures 8 and 9 show cross plots of the mass flow rate at various values of quality and test section pressure using an average value of the mass flow rate from Figures 4-7. Test section pressures ranged from 60 psia to 20 psia. Quality ranged from slight superheat to a low quality of 0.34. Pressure ratios ranged from 0.08 to 0.70, which covered the range of critical flow as well as somewhat into the region where mass flow rate is a function of pressure difference across the orifice.

From Figures 4-7, the mass flow rate of the steam is seen to decrease with an increase in pressure ratio except for dry steam. A probable explanation for this phenomenon is that a boundary layer of all or nearly all liquid had formed on the inlet of the orifice. Since the water was very probably not traveling at sonic velocity (ca 5000 ft/sec), any pressure change down stream of the orifice would be transmitted in the surface film of water as it passed the orifice. An increase in back pressure could then slow down this water layer, allowing less through and thickening

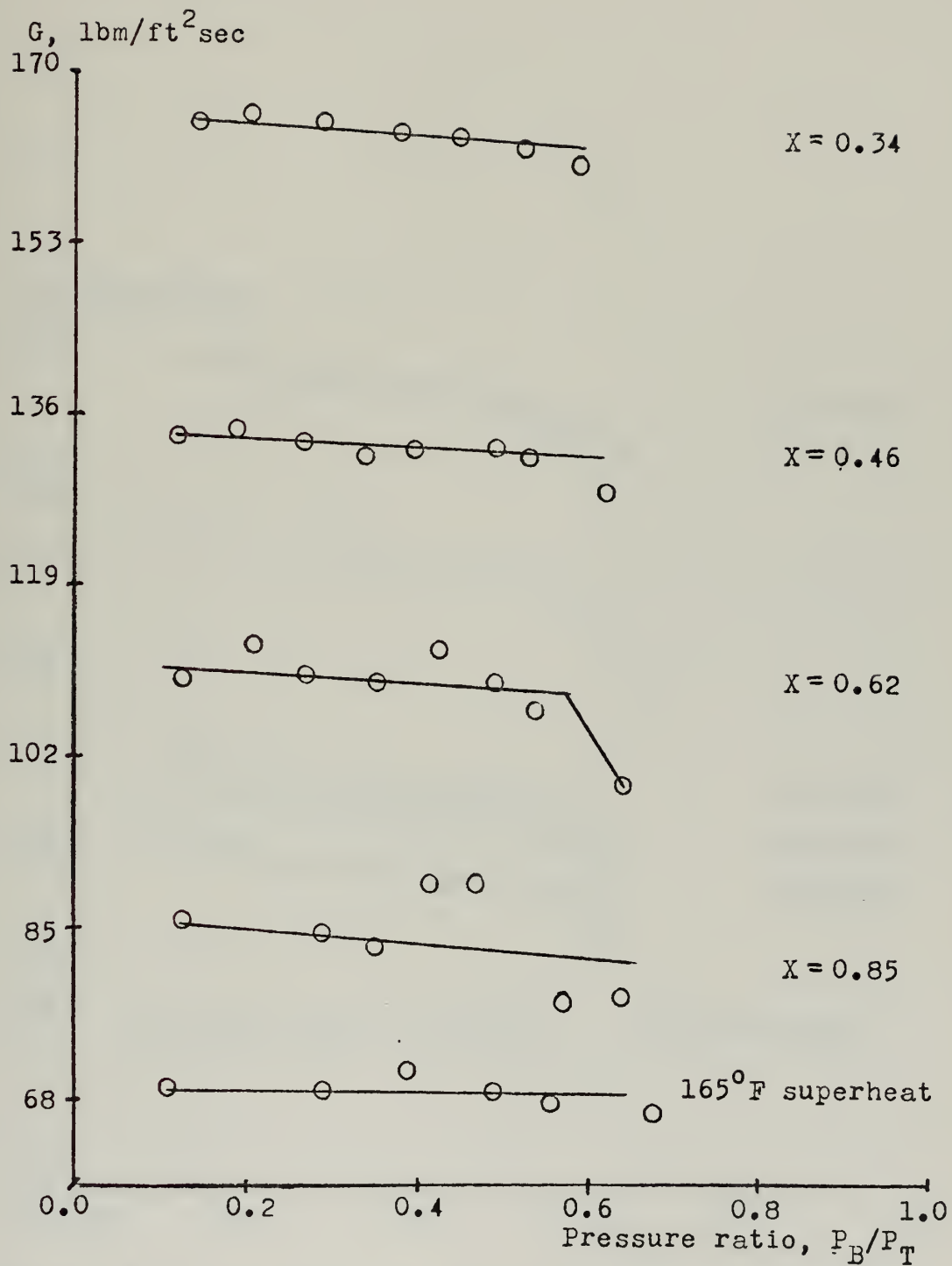


Fig. 4 Mass flow rate as a function of pressure ratio at 5 qualities. $P_T = 20$ psia

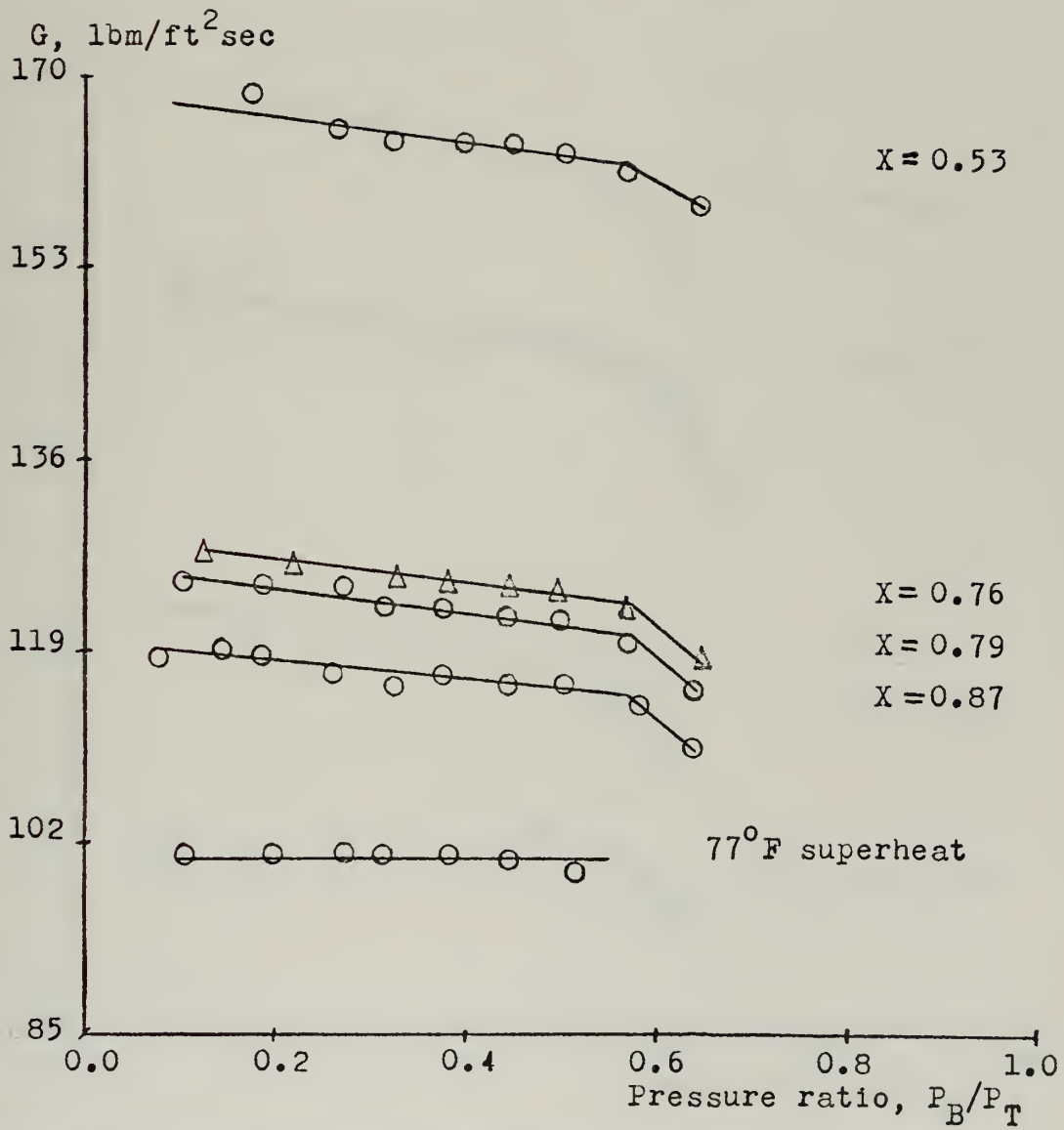


Fig. 5 Mass flow rate as a function of pressure ratio, at 5 qualities. $P_T = 29.3$ psia

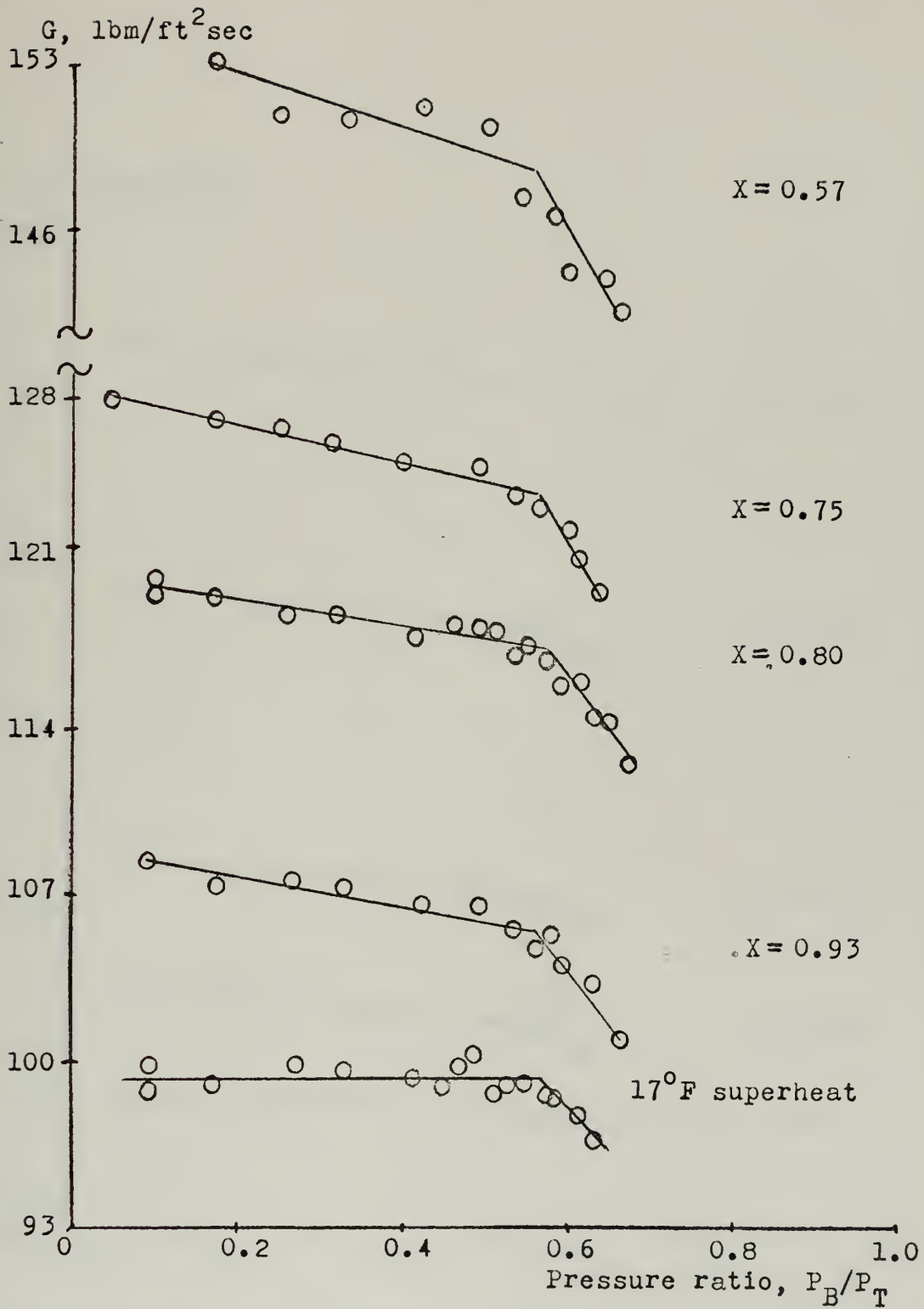


Fig. 6 Mass flow rate as a function of pressure ratio at 5 qualities. $P_T = 45$ psia

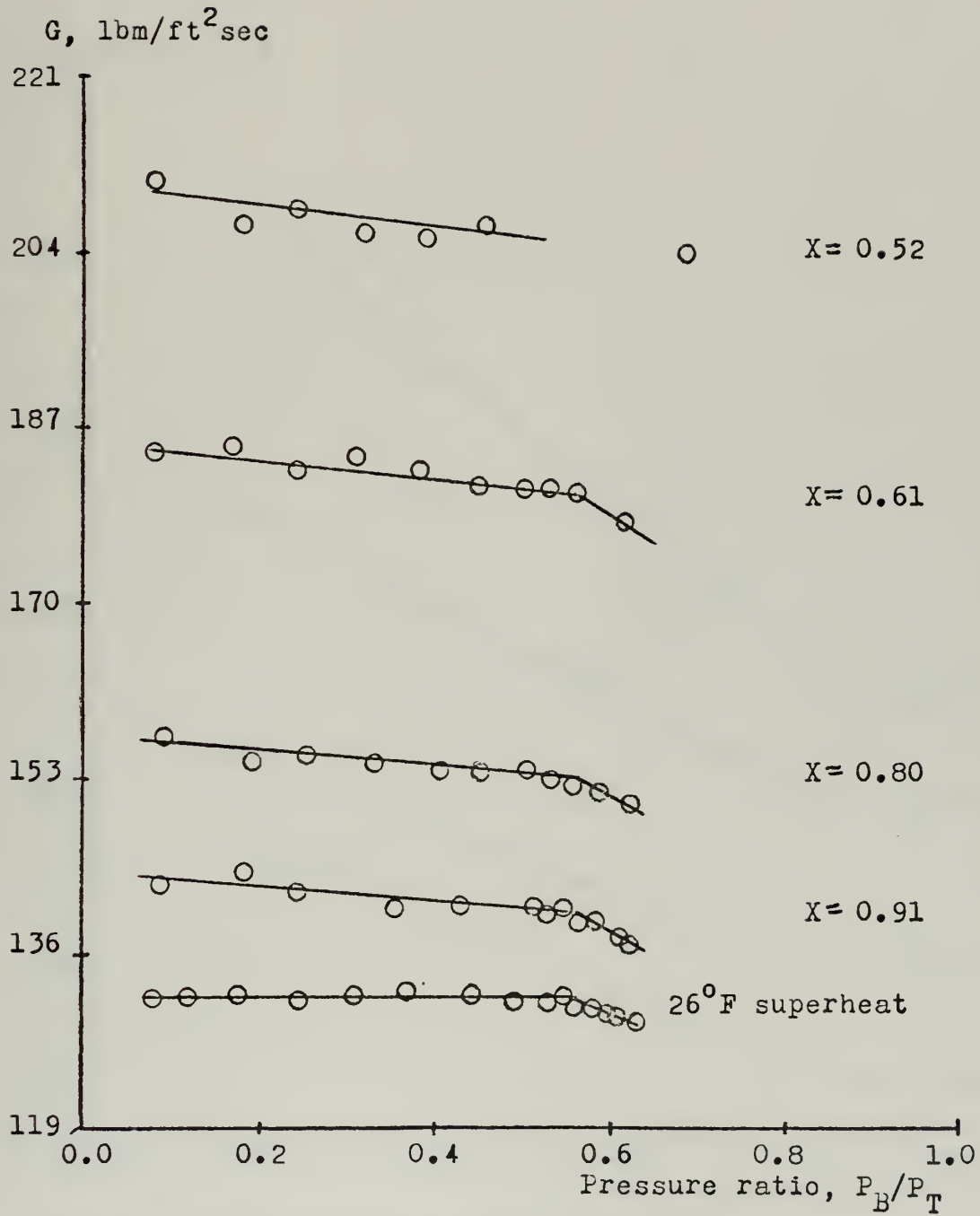


Fig. 7 Mass flow rate as a function of pressure ratio at five qualities. $P_T = 60$ psia

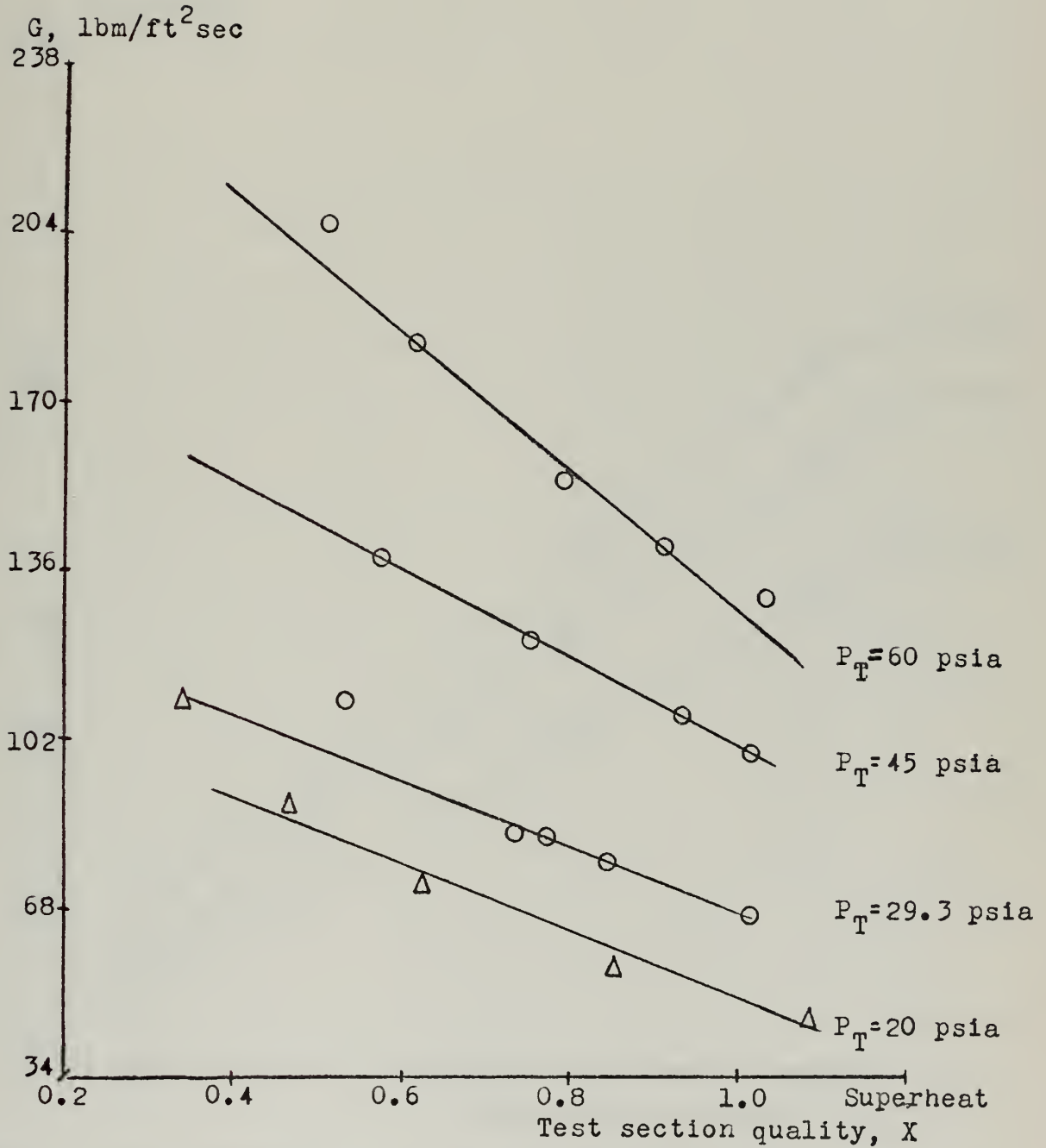


Fig. 8 Mass flow rate as a function of test section quality for 4 test section pressures. $P_B = 0.30$

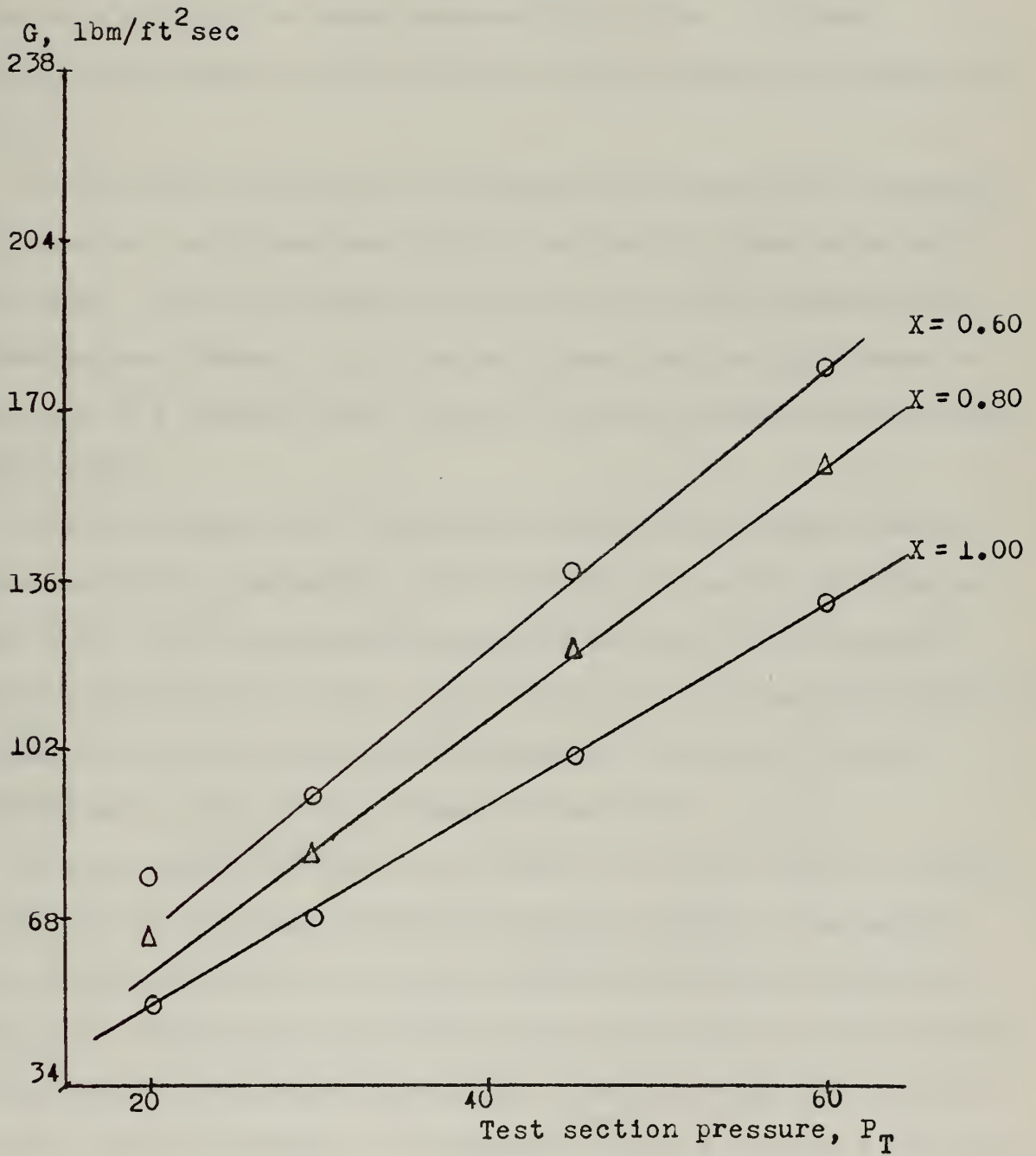


Fig. 9 Mass flow rate as a function of test section pressure at 3 qualities. $P_B = 0.30$

the boundary layer. Since the orifice was quite small any change in boundary layer thickness would cause a significant change in cross sectional area available for vapor-suspended liquid flow. (A liquid boundary layer thickness equal to 0.0015 inch would reduce the orifice area by 2%.)

To corroborate the effect of increasing test pressure with increasing back pressure, run #51 was made with the test section bypass valve left at one setting. Figure 10 shows the increase of test section pressure with increasing back pressure. As in the tests where test section pressure is maintained at a constant value, the mass flow rate decreases with increasing back pressure.

As seen in Figures 4-7, the critical flow curve approaches a knee at a pressure ratio approximately 0.58. This value is the critical pressure ratio (P_c). This value remained essentially constant over the range of qualities and pressures tested. The value 0.58 is consistent with Steltz's (5) analysis for wet steam, but is somewhat high for dry steam. The most widely accepted value for dry saturated steam is 0.54. (5,8)

Of some technical interest is the value of the slip ratio, K , defined as the ratio of the liquid velocity to the vapor velocity. One possible value of the slip ratio is $K = \rho_g / \rho_l$, called the "pressure" value of the slip ratio. The pressure value of the slip ratio is derived from comparing the velocity of pure liquid to pure vapor if driven through the orifice by the same pressure difference. The pressure value of K is about 0.001 and it should remain constant regardless of quality. In fact, the calculated results of the slip ratio are on the order of 0.02. It is interesting to note that these values of K are approximately the same as DeJong (9)

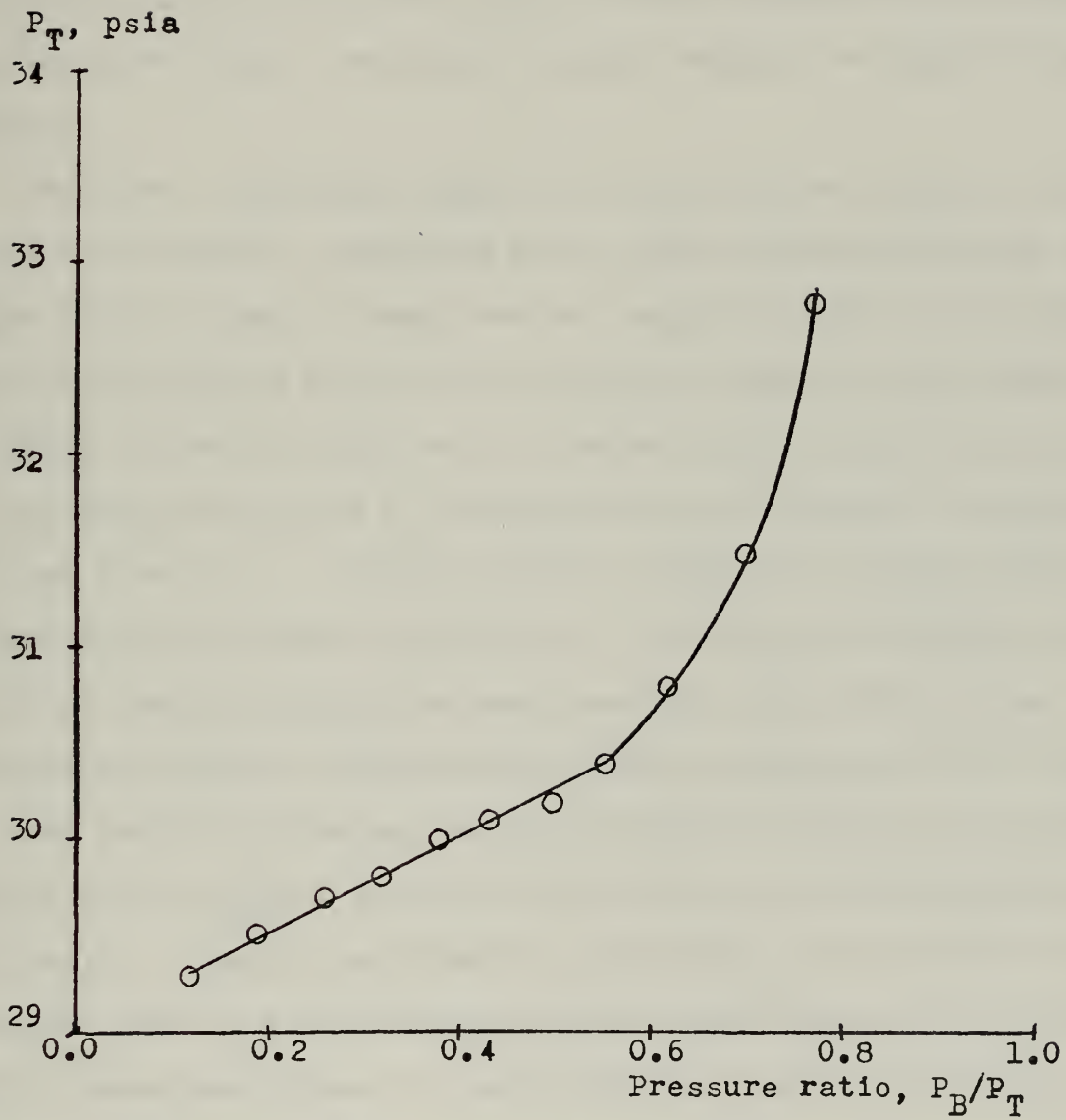


Fig. 10 Effect of back pressure on test section pressure with test section bypass opening fixed. $X = 0.80$

obtained in boiling water (very low quality) mixtures. Undoubtedly the reason for the higher values of K attained experimentally is the effect of aerodynamic drag on the droplets, just as DeJong surmised that aerodynamic drag caused the drastic reduction in acoustic velocity in liquid-continuous mixtures.

There was a substantial change in the slip ratio with change of quality at one test pressure. Considering that in vapor continuous mixtures, the volume ratio of vapor to liquid does not change substantially with change in quality, the value of K would not be expected to change for this reason. The change in the slip ratio could be reduced and the absolute value reduced by increasing the value of P_c , although there is no evidence to substantiate a higher value of P_c . Plotting K against quality for increasing degree of equilibrium in the orifice throat with $P_c = 0.58$ (Fig. 11) produced a flat curve for isentropic equilibrium conditions but a steep curve for non-equilibrium conditions representing no further condensation in the orifice.

One possible explanation of the calculated slip ratios is the possibility that the value of K varies from the center of the flow passage to the outside. Probably, the effect of a liquid film on the orifice wall and the development of a vapor-suspended liquid boundary layer in the orifice could produce local values of K quite different from the value of K in the center of the orifice. This effect would be expected to be more noticeable with small orifices.

As mentioned in the preceding chapter, some difficulty was encountered in trying to match vapor velocity with the available area. Besides allowing the value of the compressibility factor to change, variations were also tried on the assumed vapor velocity. For the initial approach, the acoustic

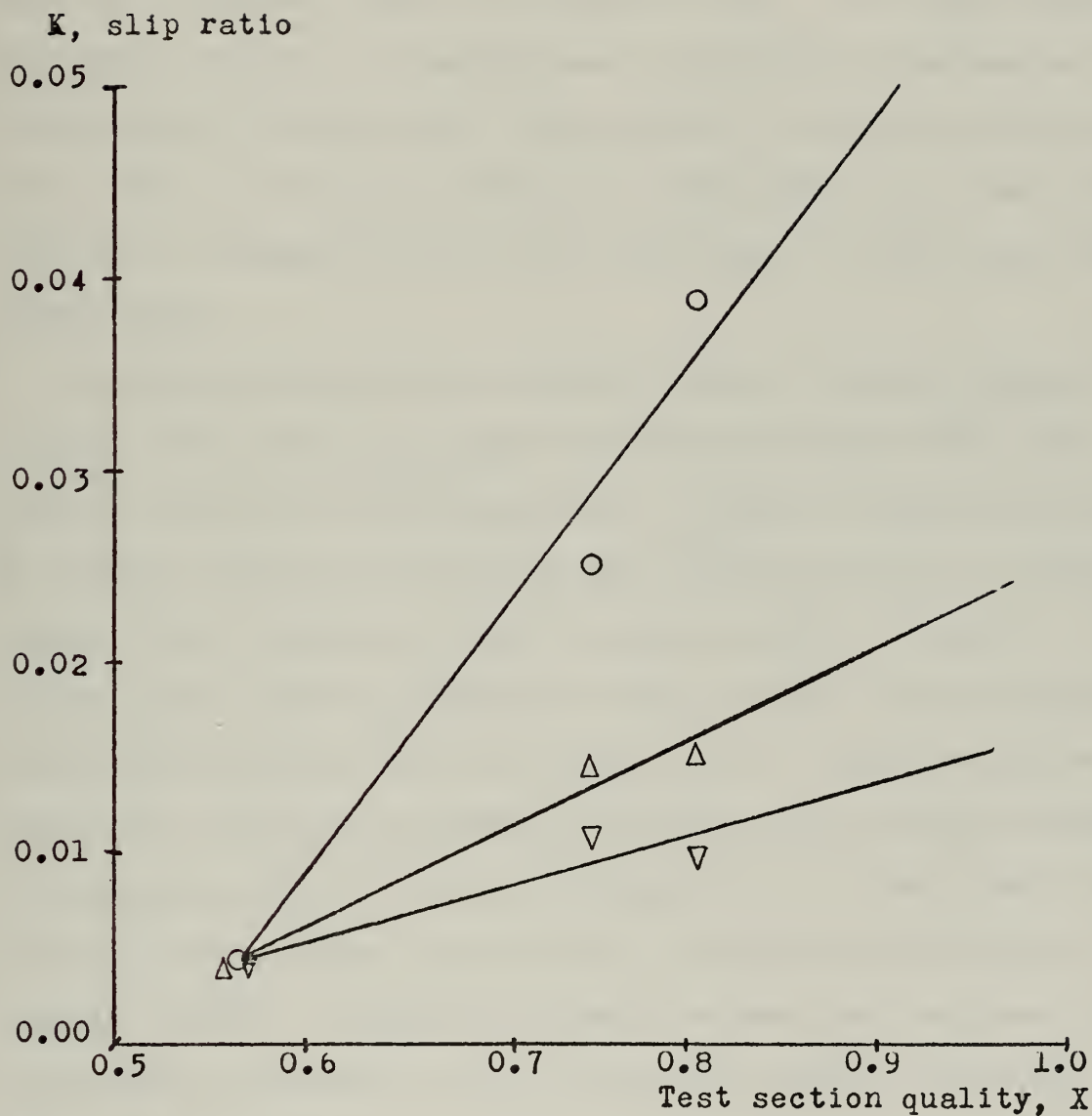


Fig. 11 Change of slip ratio as a function of test section quality for various degrees of equilibrium.

- \circ $X = \text{constant}$
- ∇ $S(\text{entropy}) = \text{constant}$
- Δ Midpoint

velocity of Trapp's data was used. The area required by the vapor portion of the flow was greater than the available area in all but the 60 psia tests for the highest quality wet steam (0.9 plus). As a second approach, the acoustic velocity of saturated vapor as determined by England was used for any quality less than unity. This approach resolved the problem at 45 psia as well as at 60 psia, although the unused margin was so small that there was no allowance for any orifice coefficient or a more slowly moving boundary layer.

The critical velocity was calculated using the specific volumes of the vapor and liquid in proportion to their amount in the whole stream. This gave results that were approximately 7% high for superheated steam. The critical velocity also decreased with decreasing quality in a pattern similar to that obtained by Trapp but somewhat steeper. (Fig. 12) The fallacy in this type of calculation is that it assumes that the vapor and liquid are traveling at the same velocity ($K = 1$). Indeed, as has been shown by the results of experiment, as well as in the literature reviewed, it is very difficult with presently available data to clearly define any velocity, whether acoustic vapor velocity, suspended liquid velocity, or critical velocity. It appears that mass flow rates must continued to be experimentally determined for use in developing empirical predictions.

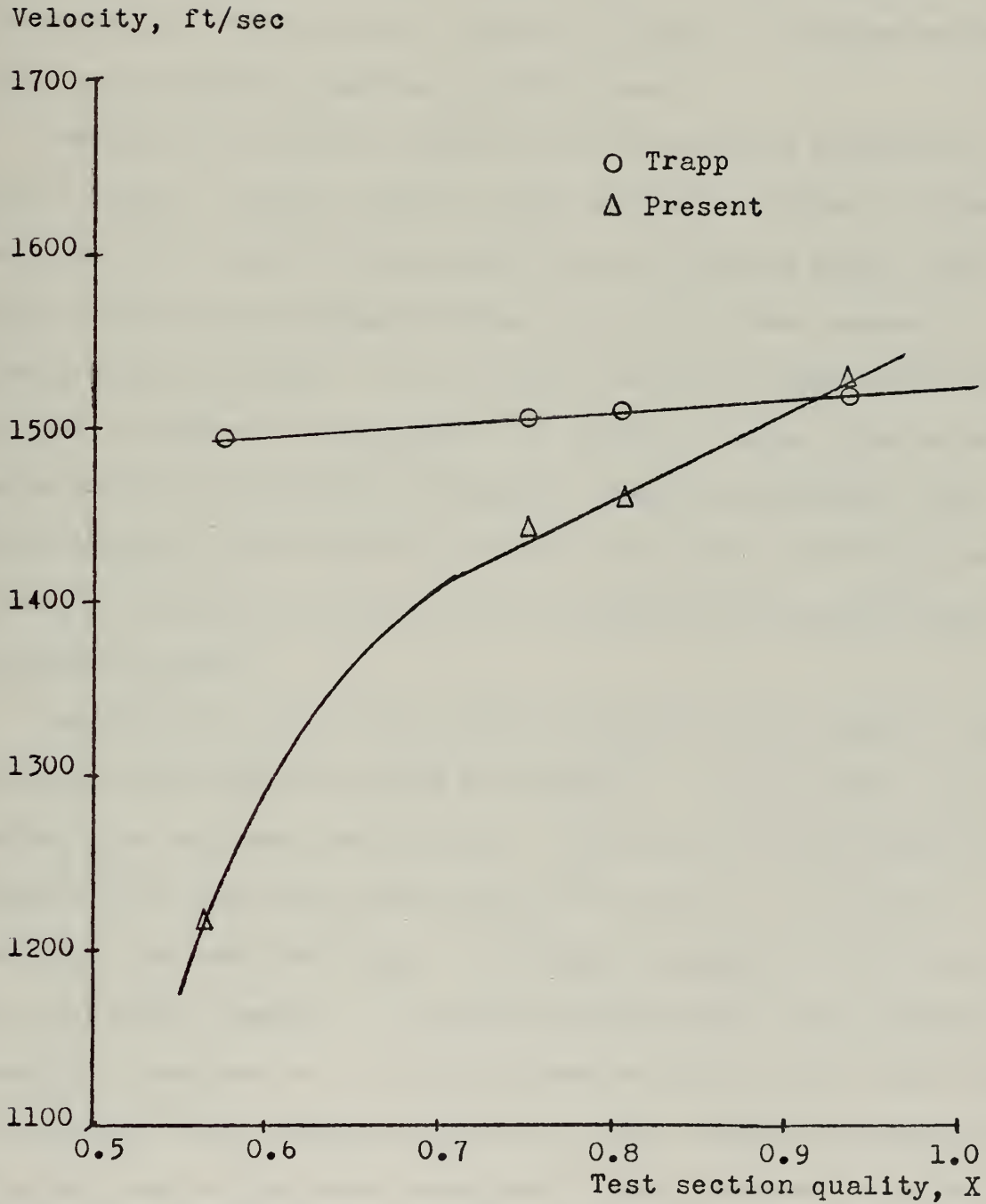


Fig. 12 Comparison of acoustic vapor velocity and average velocity of whole stream. $P_T = 45$ psia, $P_{throat} = 26.1$ psia

Recommendations

Several faults and insufficiencies are apparent in the apparatus as constructed for this experiment. Schemes to remedy the problems are presented below with the objectives desired by each.

Foremost it is desired to get away from the problems presented by small orifices. The test orifice in this case was 0.30 inches in diameter, the largest that could be accommodated with the installed supply steam nozzle which was a converging-diverging type with a throat diameter of approximately 0.25 inches. If one assumes the liquid boundary layer does not grow in thickness simply because the orifice is larger, then the error introduced by the liquid layer thickness is reduced accordingly. The vapor-suspended liquid boundary layer will also occupy a relatively smaller portion of the orifice since this type of boundary layer generally grows in a parabolic fashion.

Extending the range of test section pressures is also desired. This particular experiment was limited by a maximum of 165 psig supply pressure and the relatively small steam nozzle. Considering that there must be a pressure ratio across the steam nozzle of at least 2 to 1 in order to accelerate the steam for mixing, the principle parameter in this case is the steam supply pressure. It should be expected that results become increasingly consistent as test section pressure rises, since a small pressure change at high pressures results in a small temperature change, whereas in the range of the present experiment, a small pressure change meant a significant temperature change. Another interesting facet of higher pressures is the continuing decrease in the compressibility factor along the line of dry saturated states of steam.

An accurate determination of the slip ratio would give true values of the vapor and liquid velocities in critical flow. In order to do this, some means of controlling the flow pattern must be devised. Instead of using the water spray nozzle to provide mixing water, a ring of water bleed jets could be mounted in the test section pipe to feed water along the side of the pipe only, leaving a dry annulus. Reference 10 describes a device for measuring the film thickness. Similarly, a ring of suction jets to bleed off water condensed on the test section pipe could be mounted just forward of the test orifice. Or the test section pipe and orifice could be heated so as to boil off the film, leaving a dry outer ring and hopefully dry the orifice walls. As noted in Reference 10, caution must be used with this method to avoid geometric distortions due to high temperatures.

Visual observations could be made using the glass windows similar to those installed in this experiment. By installing the windows downstream of the orifice and extending the orifice downstream, the presence of a condensation shock for high qualities might be observed with a Schlieren apparatus. It is doubtful that a Pitot probe could be used to locate the shock since the presence of the probe would initiate its own shock.

For comparisons sake, experiments could be performed on square and sharp edged orifices using an appropriate experimental arrangement as described above.

Bibliography

1. Reynolds, Osborne, Papers on Mechanical and Physical Subjects, Vol. II, Cambridge, University Press, Cambridge, Massachusetts, (1901).
2. Henry, R. E. and Fauske, H. K., The Two-Phase Critical Flow of One Component Mixtures in Nozzles, Orifices, and Short Tubes, ASME Paper No. 70-WA/HT-5, (1970).
3. England, W. G., An Experimental Determination of the Acoustic Velocity in Wet and Slightly Superheated Steam, M.S. Thesis, University of Washington, Seattle, (1964).
4. Trapp, O. E., An Experimental Determination of the Acoustic Velocity in Wet Steam, M.S. Thesis, University of Washington, Seattle, (1965).
5. Steltz, W. G., The Critical and Two-Phase Flow of Steam, ASME Journal of Engineering for Power, New York, (April, 1961).
6. Shapiro, A. H., The Dynamics and Thermodynamics of Compressible Fluid Flow, Vol. I, Ronald Press, New York, (1953).
7. Fluid Meters, ASME, New York, (1937).
8. Steam, Its Generation and Use, The Babcock and Wilcox Co., New York, (1963).
9. Dejong, V. J., and Firey, J. C., "Effect of Slip and Phase Change on Sound Velocity in Steam-Water Mixtures and the Relation to Critical Flow," I&EC Process Design and Development, Vol. 7, American Chemical Society, New York, (July, 1968).
10. Pogson, J. T., Roberts, J. H., and Waibler, P. J., An Investigation of the Liquid Distribution in Annular Mist Flow, Journal of Heat Transfer, Vol. 92, Num. 4, ASME, New York, (1970).
11. Keenan, J. H. and Keyes, F. G., Thermodynamic Properties of Steam, John Wiley & Sons, INC. New York, (1936).

Appendix A
Calculations

The steam properties used in the experiment were taken from Thermodynamic Properties of Steam by J. H. Keenan and F. G. Keyes. (11)

The primary calculation in this experiment is the determination of test section quality. This is done by using an energy balance: Δ Energy of Steam = Δ Energy of Cooling Water. Thus:

$$\dot{m}_s h_{si} + \dot{m}_w h_{wi} = \dot{m}_s h_{so} + \dot{m}_w h_{wo}$$

$$h_{si} = \frac{\dot{m}_w}{\dot{m}_s} (h_{wo} - h_{wi}) + \frac{\dot{m}_s}{\dot{m}_s} h_{so}$$

$$h_{si} = \frac{\dot{m}_w}{\dot{m}_s} (c_p (T_o - T_i)) + (T_{con} - 32)$$

and

$$X = \frac{h_{si} - h_{fT}}{h_{fgT}}$$

From run #34, $P_T = 45$ psia:

$$\dot{m}_w = 135.5 \text{ lbm/min}$$

$$C_p = 1.0 \text{ BTU/lbm}^\circ\text{F}$$

$$\dot{m}_s = (7.059/2) \text{ lbm/min}$$

$$h_{fT} = 234.4 \text{ BTU/lbm}$$

$$T_o = 87.5^\circ\text{F}$$

$$h_{fgT} = 928.6 \text{ BTU/lbm}$$

$$T_i = 63.3^\circ\text{F}$$

$$T_{con} = 109^\circ\text{F}$$

$$h_{si} = \frac{135.5 \text{ lbm/min}}{(7.059/2) \text{ lbm/min}} (1.0 (87.5^{\circ}\text{F} - 63.3^{\circ}\text{F})) + (109 - 32)$$

$$h_{si} = 1004 \text{ BTU/lbm}$$

$$X = \frac{1004 \text{ BTU/lbm} - 243.4 \text{ BTU/lbm}}{928.6 \text{ BTU/lbm}} = 0.811$$

Note that this is the test section quality, not the quality in the throat of the orifice. In most cases ($X > 0.5$), the quality in the throat will be somewhat less, depending on the degree of reversibility of the expansion process and the degree of equilibrium condensation.

The pressure ratios were calculated from the relative heights of the mercury manometer legs and the test section pressure as read from the Bourdon gage. There was 48 inches of water above the zero mark on the left (open to atmosphere) and 49 inches of water above the zero mark on the right leg (open to back pressure).

$$\text{Pressure}_{\text{left}} = \text{Pressure}_{\text{right}}$$

$$P_1 = + 29.9 \quad \text{inches Hg (atmosphere)}$$

$$+ 48/13.6 \quad \text{inches Hg (water)}$$

$$+ -1/13.6 \quad \text{inches Hg (water for displaced mercury)}$$

$$+ 1 \quad \text{inches Hg (displacement of mercury)}$$

$$P_r = + 49/13.6 \quad \text{inches Hg (water)}$$

$$+ -r/13.6 \quad \text{inches Hg (water for displaced mercury)}$$

$$+ r \quad \text{inches Hg (displacement of mercury)}$$

$$\text{Pressure}_{\text{back}} = P_1 - P_r$$

$$P_B = (29.9 + \frac{48-1}{13.6} + 1) \text{ in Hg} - (\frac{49-r}{13.6} + r) \text{ in Hg}$$

$$P_B = (29.9 + \frac{12.6(1-r)-1}{13.6}) \text{ in Hg}$$

$$\text{Pressure Ratio} = \frac{P_B}{P_T}$$

$$\text{Pressure Ratio} = \frac{(29.9 + \frac{12.6(1-r)-1}{13.6}) \text{ in Hg} \times (0.4912 \text{ psi/in Hg})}{P_T \text{ psi}}$$

$$\text{Let } l = -10.9 \text{ in Hg}$$

$$r = 11.7 \text{ in Hg}$$

$$P_T = 45 \text{ psia}$$

$$\text{Pressure Ratio} = 0.100$$

The slip ratio was calculated by assuming a velocity for the vapor phase, determining the orifice area required by the vapor phase, then using the remaining area to find the liquid velocity. The slip ratio was then the ratio of the liquid velocity to vapor velocity. The compressibility factor, Z , was included in these calculations.

$$Z = \frac{RT}{Pv}$$

$$R = \frac{1545 \text{ ft.lbf/lbm.mole } ^\circ\text{R}}{18.0 \text{ lbm/lbm.mole}}$$

$$T = 702^\circ\text{R}$$

$$P = 26 \text{ psia} = 3744 \text{ psfa}$$

$$v = 15.715 \text{ ft}^3/\text{lbm}$$

$$Z = \frac{\frac{1545 \text{ ft} \cdot \text{lb}_f / \text{lbm} \cdot \text{mole}^{\circ}\text{R}}{18.0 \text{ lbm} / \text{lbm} \cdot \text{mole}} \times 702^{\circ}\text{R}}{3744 \text{ lb}_f / \text{ft}^2 \times 15.715 \text{ ft}^3 / \text{lbm}}$$

$$Z = 0.9773$$

$$\dot{m} = \frac{A_g V_g}{v_g} + \frac{A_f V_f}{v_f}$$

$$X = \frac{\dot{m}_g}{\dot{m}_t}$$

$$X = \frac{A_g V_g}{v_g} / \dot{m}_t$$

$$A_g = \frac{X \dot{m}_t v_g}{V_g}$$

Including compressibility;

$$A_g = \frac{X \dot{m}_t v_g Z}{V_g}$$

$$A_f = A_t - A_g$$

$$v_f = \frac{\dot{m}_t - \frac{A_g V_g}{v_g}}{A_f / v_f}$$

$$v_f = \frac{v_f \dot{m}_t (1 - X)}{A_f}$$

$$K = \frac{v_f}{v_g}$$

Values are derived from run #34 with $P_T = 45$ psia, $P_c = 0.58$, so that pressure in the throat equals 26.1 psia.

Let $X = 0.805$

$$m_t = 3.520 \text{ lbm/min.}$$

$$v_f = 0.01694 \text{ ft}^3/\text{lbm}$$

$$v = 15.658 \text{ ft}^3/\text{lbm}$$

$$A_t = 0.0004906 \text{ ft}^2$$

$$V_g = 1511 \text{ ft/sec}$$

$$A_g = \frac{0.805 \times \frac{3.520 \text{ lbm}}{60 \text{ sec}} \times (1 - 0.805)}{0.0000123 \text{ ft}^2}$$

$$V_f = 15.7 \text{ ft/sec}$$

$$K = \frac{15.7 \text{ ft/sec}}{1511 \text{ ft/sec}} = 0.010$$

The critical velocity is derived from the specific volumes of the liquid and vapor phases passing through the orifice. Again the compressibility is included.

$$V_c = \frac{m_t}{A_t} (X v_g Z + (1 - X) v_f)$$

Using the values from the previous example:

$$V_c = \frac{\frac{3.520 \text{ lbm}}{60 \text{ sec}}}{0.0004906 \text{ ft}^2} \times (0.805 \times 15.658 \frac{\text{ft}^3}{\text{sec}} \times 0.9773 + (1 - 0.805) \times 0.01694 \frac{\text{ft}^3}{\text{lbm}})$$

$$V_c = 1474 \text{ ft/sec}$$

Appendix B

Recorded Data

RUN #33
15 AUG 70
DRY RUN

$P_{\text{act}} = 5.3 \text{ psig}$
 $T_{\text{air}} = 296^\circ \text{F}$

$P_s = 169 \text{ psig}$.300 orifice
1 min cooling
2 min sample

Pb/Pt	Pt	L	R	V	TIN	Tout	Tout	Cal	Time	Wt	X	Vol
.043	30.3	-11.1	+12.2	24.5	63.0	86.7	107	5.815	106 ²	243 ¹	100.5	1406
					63.0	86.8	108	5.836		136 ³	102.3	1372
.174	30.3	-7.1	+8.1	24.5	63.0	86.7	108	5.854			101.4	1377
.270	30.3	-2.5	+3.2	24.5	63.0	86.8	108	5.871			101.1	1405
.321	30.3	+0.4	+0.2	24.5	63.0	86.8	108	5.885	106 ⁴	244 ²	101.4	1403
.414	30.3	+4.5	-4.1	24.5	63.0	86.7	108	5.859		137 ³	101.3	1378
.453	30.3	+6.4	-6.1	24.5	62.9	86.6	108	5.836			101.8	1372
.471	30.3	+7.3	-7.0	24.4	62.9	86.9	108	5.895			102.1	1406
.489	30.3	+8.2	-7.9	24.3	62.8	86.8	108	5.925			101.5	1413
.507	30.3	+9.1	-8.8	24.3	62.7	86.7	108	5.916			102.7	1388
.523	30.3	+10.0	-9.7	24.3	62.8	86.4	107	5.897	106 ⁵	243 ²	101.0	1375
.554	30.3	+11.4	-11.1	24.3	62.8	86.5	107	5.847		137 ⁴	101.4	1375
.576	30.3	+12.5	-12.2	24.2	62.9	86.6	107	5.824			101.9	1389
.588	30.3	+13.1	-12.8	24.2	63.0	86.5	108	5.811			101.1	1386
.612	30.3	+14.3	-14.0	24.2	63.1	86.7	108	5.771			102.6	1377
.628	30.3	+15.1	-14.8	24.2	63.1	86.5	108	5.711			102.1	1362
.648	30.3	+16.1	-15.8	24.2	63.1	86.3	107	5.692			102.1	1358
Average - 10 points												
.616	30.3	+14.5	-14.2	24.2	63.0	86.4	107	5.762			101.7	1375
.594	30.3	+13.4	-13.1	24.2	62.9	86.4	108	5.813			101.3	1397
.568	30.3	+12.1	-11.8	24.2	63.0	86.6	108	5.872			101.5	1401
.531	30.3	+10.3	-9.9	24.2	63.0	86.7	108	5.804			102.4	1385
ave												(.1017)

ave of pts 1-10 = 1400.0

Run # 34
16 SEP 70

$P_{a1} = 2.7 \text{ psig}$ $P_w = 152 \text{ psig}$
 $T_{a1} = 222^\circ \text{F}$ $T_w = 196^\circ \text{F}$
 $P_s = 113 \text{ psig}$

300 orifice
40 GPH w/ PL insert
1 min cooling
2 min sample

P_{a1}	P_s	L	R	V	T_{a1}	T_{a2}	T_{a3}	Q_{a1}	T_{a4}	W	X
.100	30.3	-10.4	+11.7	24.2	63.3	87.5	109	7.059	105%	240	81.1
					63.3	87.3	109	7.017	134%		81.4
.171	30.3	-7.4	+8.1	24.2	63.3	87.3	110	7.021			81.6
.260	30.3	-3.0	+3.7	24.2	63.3	87.2	109	6.951			82.1
					63.3	87.2	109	6.972			81.8
.322	30.3	0.0	+0.5	24.2	63.4	87.2	109	6.969			81.4
.410	30.3	+4.2	-4.0	24.2	63.4	87.1	108	6.923			81.5
.453	30.3	+6.2	-6.3	24.2	63.3	86.8	108	6.948	106%	242	80.3
.491	30.3	+8.2	-8.1	24.2	63.3	86.9	109	6.946	136%		80.8
.512	30.3	+9.2	-7.2	24.2	63.4	86.9	108	6.935			80.5
.535	30.3	+10.3	-10.3	24.1	63.4	87.0	108	6.878			81.7
.544	30.3	+11.0	-11.0	24.1	63.4	86.9	107	6.906			80.8
.564	30.3	+12.0	-12.0	24.1	63.4	86.5	107	6.863			79.7
.591	30.3	+13.1	-13.1	24.1	63.4	86.5	107	6.814			80.4
.614	30.3	+14.2	-14.3	24.1	63.4	86.4	107	6.819			79.9
.632	30.3	+15.2	-15.2	24.1	63.4	86.2	107	6.731			80.4
.651	30.3	+16.1	-16.1	24.1	63.3	86.0	106	6.702			80.2
.676	30.3	+17.3	-17.3	24.1	63.3	85.6	105	6.618			79.6
Reverse											
.644	30.3	+16.0	-16.0	24.1	63.3	86.0	106	6.688	106%	241	80.4
.677	30.3	+14.9	-14.9	24.1	63.3	86.2	106	6.811	135%		79.5
.603	30.3	+13.7	-13.7	24.1	63.3	86.4	106	6.879			79.9
.564	30.3	+12.0	-12.0	24.1	63.3	86.4	106	6.905			79.0
.547	30.3	+10.9	-10.9	24.1	63.4	86.7	107	6.934			79.6
.529	30.3	+10.0	-10.0	24.1	63.4	86.7	107	6.920			79.8
.504	30.3	+8.8	-8.8	24.1	63.4	86.9	107	6.952			80.2
.483	30.3	+7.8	-7.7	24.1	63.4	87.0	108	6.968			80.5

ave
(0.805)

$P_{\text{air}} = 2.9 \text{ psig}$ $P_w = 150 \text{ psig}$
 $T_{\text{air}} = 223^\circ \text{ F}$ $T_w = 197^\circ \text{ F}$
 $P_s = 118 \text{ psig}$

• 300 office
17.5 GPA no 3 le PL
1 min cool (ing
2 min sample

[illegible]

RUN #37

24 SEPT 70

P_{at} = 2.8 psig T_{at} = 223°FP_w = 150 psig T_w = 296°FP_s = 115 psig V = 24.4

.300 orifice

60.0 GPH nozzle
w/PL insert
1 min cooling
2 in sample

P _{at}	P _t	L	R	T _{IN}	T _{OUT}	T _{IN}	P _{IN}	TIME	WT	X
.092	30.3	-11.4	+12.0	62.3	87.5	109	7.491	103 ³	132 ³	.755
				62.3	87.5	109	7.500	129.0		.754
.171	30.3	-7.5	+8.0	62.4	87.3	109	7.439			.751
.252	30.3	-3.5	+3.9	62.4	87.3	109	7.431			.752
.316	30.3	-0.5	+0.6	62.4	87.1	109	7.303			
				62.4	87.2	109	7.416			.750
				62.4	87.2	109	7.391			.753
.402	30.3	+3.7	-3.7	62.4	87.0	109	7.336			.752
.496	30.3	+8.3	-8.5	62.4	86.8	108	7.330	105 ³	234 ³	.755
.529	30.3	+10.2	-10.3	62.5	86.7	108	7.272	128.75		.741
.561	30.3	+11.7	-11.9	62.5	86.5	108	7.236			.741
.601	30.3	+13.5	-13.7	62.4	86.4	108	7.183			.748
.612	30.3	+14.0	-14.3	62.4	86.2	107	7.106			.749
.639	30.3	+15.3	-15.7	62.5	86.0	107	7.021			.749
				62.4	86.1	106	7.033			.751

RECHECK

RECHECK

.599	30.3	+13.9	-13.6	62.4	86.5	107	7.209	105 ³	234 ³	.747
.574	30.3	+12.1	-12.4	62.4	86.8	108	7.282	129.0		.751
.542	30.3	+10.5	-10.8	62.5	86.9	109	7.291			.752

ave
(.750)

RUN #38
30 SEPT 70

$P_{CH} = 3.0 \text{ psig}$

$P_W = 150$

$P_s = 119$

$T_{CH} = 223^\circ\text{F}$

$T_W = 196^\circ\text{F}$

$V = 241$

.300 c/fce

100 GPM PLP nozzle

1 min cooling

P_B/P_T	P_T	L	R	T_{IN}	T_{OUT}	T_{CAN}	CON	THICK	WT	X
.079	30.3	-12.1	+12.5	61.9	84.2	106	8.690	106 ³	250 ³	52.2
				61.9	84.2	106	7.700		142.0	52.2
.168	30.3	-7.7	+8.1	62.0	84.2	106	8.600			57.3
				62.0	84.1	106	8.740			57.5
.249	30.3	-3.8	+4.0	62.0	84.0	106	8.674			57.7
				62.1	84.0	106	8.869			57.4
.330	30.3	+2.2	-0.1	62.2	84.3	105	8.571	105 ²	247 ³	57.9
.421	30.3	+4.5	-4.8	62.2	84.1	106	8.679		142.2	57.1
.502	30.3	+8.6	-8.8	62.3	84.2	106	8.848			57.6
.538	30.3	+10.3	-10.6	62.3	84.0	105	8.691			58.1
.580	30.3	+12.4	-12.7	62.3	83.7	104	8.607			57.7
				62.3	83.6	109	8.639			57.1
.602	30.3	+13.5	-13.8	62.3	83.5	103	8.500			57.9
.641	30.3	+15.5	-15.7	62.2	83.0	102	8.487			56.4
.661	30.3	+16.5	-16.7	62.2	82.7	102	8.409			56.0
.702	30.3	+18.5	-18.7	62.3	82.4	101	8.075			57.5
				62.3	82.2	101	8.105			56.4
<hr/>										
.577	30.3	+13.2	-13.6	62.2	83.4	103	8.613			56.8
.325	30.3	+11.6	-11.8	62.2	83.5	104	8.719	105 ²	248 ³	56.4
.456	30.3	+6.2	-6.6	62.2	83.8	105	8.774		142.5	57.0
.327	30.3	0	0	62.2	84.2	106	8.852			57.9
.201	30.3	-6.1	+6.4	62.3	84.4	107	8.890			58.0
.102	30.3	-11.0	+11.4	62.3	84.6	108	8.965			58.2

(.573)

Run #40
12 OCT 70

$P_{\text{CAL}} = 5.5 \text{ psig}$ $P_2 = 169 \text{ psig}$
 $T_{\text{CAL}} = 296^\circ\text{F}$ $V = 23.5$
 $T_{\text{CON}} = 144$

0300 nozzle
DRY RUN
1 min cooling
2 min sample

[illegible]

RUN #42

16 OCT 70

page 1 of 2

 $P_{\text{air}} = 4.9 \text{ psig}$ $T_{\text{air}} = 222^\circ\text{F}$ $P_{\text{w}} = 151 \text{ psig}$ $T_{\text{w}} = 198^\circ\text{F}$ $P_s = 159 \text{ psig}$ $V = 23.5$ $T_s = 142^\circ\text{F}$

.300 orific

100 GPH w/PLP

1 min cooling

1 min sample

Pb/Pt	P _r	L	R	T _{in}	T _{out}	T _{cen}	Cen	T _{irr}	Wt	X
.082	45.3	-11.0	+10.7	58.8	77.9	114	594	107 ²	317 ²	51.6

6.139 211²

6.206

6.251

~~6.343~~

6.179

6.291 (ave x 2)

6.145 (12.398)

.155	45.3	-4.2	+3.6	58.8	77.6	114	5.166	108 ²	319 ²	51.5
------	------	------	------	------	------	-----	-------	------------------	------------------	------

6.007 210²

6.178

6.153

6.161 (ave x 2)

5.951 (12.155)

.242	45.3	-0.9	+6.2	58.6	77.5	114	6.179	106 ²	319 ²	51.5
------	------	------	------	------	------	-----	-------	------------------	------------------	------

6.023 213²

6.091

6.065

6.162

~~6.273~~ (ave x 2)

6.176 (12.232)

.316	45.3	+4.2	-5.0	58.7	77.5	114	6.149	106 ²	320 ²	52.7
------	------	------	------	------	------	-----	-------	------------------	------------------	------

6.012 213²

6.003

6.083

5.923 (ave x 2)

6.136 (12.100)

.395	45.3	+9.3	-10.4	58.7	77.3	114	6.276	106 ²	219 ²	52.0
------	------	------	-------	------	------	-----	------------------	------------------	------------------	------

5.746 213²

6.175

6.124

6.057

6.039

6.031

5.157 (ave x 2)

6.000 (12.082)

Run #42
page 2 of 2

PB/IT	P _T	L	R	T _{IN}	T _{OUT}	T _{NEW}	PERJ	Time	WIT	X
.459	45.3	13.7	-14.4	58.9	77.4	113	6.257	1063	2182	50.6

~~6.220~~ 2113

6.144

6.051

6.121

6.046

6.055 (ave x 2)

6.050 12.156

.685	45.3	18.3	-19.0	58.9	77.3	113	6.229	1061	3181	51.3
------	------	------	-------	------	------	-----	------------------	------	------	------

5.930 3122

5.793

6.032

~~6.214~~

~~6.125~~

~~5.832~~

6.056

5.992 ave x 2

5.983 (11.995)

ave

.516

RUN #43
20 OCT 70

$P_{01} = 4.9 \text{ psig}$ $P_{02} = 156 \text{ psig}$ $.300 \text{ n/a}$
 $T_{01} = 222^\circ\text{F}$ $T_{02} = 197^\circ\text{F}$ 24.6 GPM PL
 $P_s = 155 \text{ psig}$ $V = 22.5$ 1 min cooling

P_{01}/P_T	P_T	L	R	T_{IN}	T_{OUT}	T_{CEN}	CEN	TIME	WT
.024	45.3	-10.5	+10.2	58.0	77.5	117	8.385	105° 323'	
							8.905	218'	.915
.182	45.3	-4.4	+4.0	58.0	77.6	118	8.338		.930
							8.352		
.244	45.3	-0.5	-0.2	58.0	77.4	118	8.346		
							8.269		.923
							8.300		
.357	45.3	+6.8	-7.8	57.5	76.9	116	8.280		.924
.430	45.3	+11.7	-12.6	57.8	76.8	116	8.261		.903
.513	45.3	+17.2	-18.0	57.8	77.0	115	8.242		.916
.527	45.3	+18.1	-19.0	57.8	76.9	115	8.224		.913
.544	45.3	+19.2	-20.1	57.7	76.7	115	8.257	105° 324'	.903
							8.207	219'	.909
							8.299		.914
.563	45.3	+20.6	-21.3	57.5	76.3	116	8.180		.903
.587	45.3	+22.1	-22.9	57.7	76.5	115	8.200		.899
							8.180		.901
.610	45.3	+23.7	-24.3	57.7	76.5	114	8.097		.912
.621	45.3	+24.9	-25.1	57.7	76.4	114	8.058	105° 323'	.911
								218'	
<hr/>									
.484	45.3	+15.6	-15.8	57.6	76.6	114	8.259		.901
.334	45.3	+5.7	-5.9	57.6	76.7	115	8.251		.909
.220	45.3	-1.6	+1.8	57.7	76.9	116	8.327		.906

ave
.910

RUN #44
23 Oct 70

$P_{CA} = 4.9 \text{ psig}$ $P_W = 157 \text{ psig}$ $P_S = 157 \text{ psig}$ $\phi 30 \text{ or } 1/2 \text{ in}$
 $T_{CAL} = 222^\circ\text{F}$ $T_W = 196^\circ\text{F}$ $V = 22.5 \text{ in}$ $40.0 \text{ GPH w/ } 1/2 \text{ in insert}$
 1 mm ceramic
 1 min sample

$P_{E/P}$	P_T	L	R	T_{in}	T_{out}	T_{avg}	(C_p)	Time	Wt	$C_{eff}(t)$
.089	45.3	-10.2	+10.6	57.0	79.5	116	4.624	105% 2873	.792	9.212
							4.588	1843		
.195	45.3	-3.3	+3.6	57.2	79.6	116	4.560		.801	9.078
							4.518			
.255	45.3	+0.6	-0.6	57.2	79.5	117	4.569		.792	9.133
							4.569			
.329	45.3	+5.3	-5.6	57.2	79.6	116	4.563		.800	9.090
							4.512			
							4.560			
.405	45.3	+10.4	-10.6	56.9	79.8	115	4.505		.795	9.051
							4.544			
							4.995	105% 2643		
							4.537	1793		
.459	45.3	+14.0	-14.1	56.9	80.1	116	4.533		.812	9.027
							4.494			
.506	45.3	+17.1	-17.2	56.9	79.8	116	4.529	105% 2890	.797	9.045
							4.543	1792		
							4.519			
							4.498			
.534	45.3	+18.9	-19.1	57.0	79.7	115	4.504		.792	8.994
							4.490			
.558	45.3	+20.6	-20.6	57.0	79.7	114	4.477	105% 2343	.797	8.952
							4.475	1792		
.590	45.3	+22.7	-22.7	57.1	79.6	114	4.467		.790	8.934
							4.467			
.625	45.3	+25.0	-25.0	57.1	79.5	114	4.439		.795	8.897
							4.408			
							Reverse	Reverse		
.463	45.3	+4.3	-4.4	57.1	79.8	115	4.542		.780	9.109
							4.567			
.314	45.3	+4.5	-4.5	57.0	79.9	116	4.501		.796	9.032
							4.507			
							4.570			
.170	45.3	-4.9	+5.2	57.0	79.8	117	4.562		.790	9.084
							4.522			
							ave			
							.795			

RUN #46
7 Nov 70

$P_a = 9.9 \text{ psig}$ $T_{ca} = 222^\circ\text{F}$
 $P_w = 150 \text{ psig}$ $T_w = 197^\circ\text{F}$
 $P_s = 156 \text{ psig}$ $V = 23.5 \text{ in}$

.30 of ice
 600 GPH w/PLP
 1 min cooling
 1 min sample

P ₃ /PT	P _T	L	R	T _{IN}	T _{OUT}	T _{CH}	GN	TIME	WT	X
.079	45.3	-12.8	+11.2	53.8	74.7	111	5.493	100 ³ 300 ³	.659	
							5.436	179 ³		
							5.382	(10.891)		
.170	45.3	-4.8	+5.2	53.8	74.1	111	5.435		.616	
							5.496			
							5.488			
							5.375	(10.887)		
.243	45.3	-0.2	+0.2	53.8	73.9	111	5.446		.618	
							5.327			
							5.342			
							5.382	(10.748)		
.310	45.3	+4.1	-4.3	53.8	73.9	111	5.431		.613	
							5.370			
							5.436			
							5.382	(10.809)		
.379	45.3	+8.7	-8.9	53.8	73.7	110	5.445	105 ³ 306 ³	.609	
							5.423	201 ³		
							5.271			
							5.398			
							5.332	(10.747)		
.446	45.3	+13.1	-13.3	53.8	73.6	110	5.410		.610	
							5.238			
							5.500			
							5.367			
							5.336	(10.676)		
.498	45.3	+16.5	-16.7	53.9	73.7	111	5.278		.614	
							5.353			
							5.278			
							5.368	(10.639)		
.524	45.3	+18.6	-18.8	54.0	73.8	110	5.341	105 ³ 305 ³	.613	
							5.303	200 ³		
							5.375			
							5.276	(10.658)		

Run #46

Page 2

$P_{\frac{1}{2}}$	R	L	R	TIN	TOUT	TRON	CAN	TAKE	WT	X
.561	45.3	+20.7	-20.9	53.9	73.5	110	5.359 5.277 5.292 <u>5.303</u>	(10.616)		.607
.617	45.3	+24.9	-24.6	53.9	73.5	109	5.426 5.206 5.294 5.186 <u>5.246</u>	(10.466)		.617
										(ave .613)

Run #48
11 Nov 70
 $P_T = 14.6 \text{ psig}$

$P_{CLF} = 1.6 \text{ psig}$
 $T_{CLF} = 241^\circ\text{F}$

$V = 25.9$
 $P_g = 86 \text{ psig}$

.30 oil/ice
DRY RUN
2 min cooling
3 min sample

P_{CLF}	L	R	TIN	TOUT	T-EN	LEN	T-PL	WT	X
.113	-12.4	+12.8	54.3	73.9	78	5.944	102 ³	336 ²	1.050
						5.969		233.75	
.203	-9.5	+9.9	54.3	73.5	78	6.665			
						5.950			1.026
.274	-7.1	+7.4	54.2	73.7	79	5.938			1.099
.314	-6.2	+6.0	54.2	73.7	78	5.949	(236)		1.045
.344	-3.5	+3.6	54.2	73.5	78	5.933			1.036
.447	-1.8	+1.9	54.2	73.4	78	5.911			1.036
						5.890			
.519	+0.3	-0.5	54.0	73.1	79	5.845	104 ³	342 ²	1.044
						5.839		238	

cooling water flow broke

ave
(1.041)

RUN #49

13 Nov 70

 $P_T = 14.6 \text{ psig}$ $P_{\text{air}} = 2.0 \text{ psig}$ $T_{\text{air}} = 220^\circ \text{F}$ $P_{\text{w}} = 123 \text{ psig}$ $T_{\text{w}} = 197^\circ \text{F}$ $P_s = 90 \text{ psig}$ $V = 26.4 \text{ in Hg}$

.30 or 1/2 ice

24 GPH nozzle w/PL

2 min coal

3 min sample

P_B / P_T	L	R	TIN	Tout	Tcon	CON	TARE	WT	X
.085	-13.3	+13.7	53.9	74.2	81	6.988	104 ²	331 ²	.867
						6.973	227		
.150	-11.2	+11.6	54.0	74.2	82	7.027			.856
.194	-9.8	+10.2	54.0	74.1	82	6.950			.867
.265	-7.5	+7.9	54.0	74.2	82	6.884			.878
						6.883	(226.7)		
.329	-5.5	+5.8	54.0	74.3	81	6.833			.890
.380	-3.9	+4.1	54.0	74.1	79	6.875			.871
.448	-1.7	+1.9	54.0	73.5	79	6.921			.848
						6.818			
.510	+0.2	-0.2	54.0	73.7	81	6.822	105 ²	332 ²	.860
							226.5		
.580	+2.4	-2.5	54.0	73.8	80	6.696			.883
						6.700			
.640	+4.3	-4.5	54.1	73.1	78	6.500			.869

(ave
268)

RUN #50
25 Nov 70
 $P_T = 14.6 \text{ psig}$

$P_{\text{air}} = 2.1 \text{ psig}$ $T_{\text{air}} = 218^\circ$
 $P_{\text{H}_2\text{O}} = 120 \text{ psig}$ $T_{\text{H}_2\text{O}} = 174^\circ$
 $P_g = 40 \text{ psig}$ $V = 25.8$

.30 of ice
40 GPH w/PH inlet
2 mm cooling
3 mm sample

$P_{\text{B/T}}$	L	R	$T_{\text{H}_2\text{O}}$	$T_{\text{H}_2\text{O}}$	$T_{\text{H}_2\text{O}}$	$T_{\text{H}_2\text{O}}$	$T_{\text{H}_2\text{O}}$	WT	X
.104	-12.9	+12.9	50.1	71.2	75	7.349	105'	323°	.805
						7.366	217.7		
.141	-10.1	+10.1	50.1	70.6	75	7.355			.778
.277	-7.3	+7.3	50.1	70.4	75	7.339			.770
.322	-5.9	+5.8	50.2	71.0	74	7.206			.811
						7.219			
.377	-4.2	+4.0	50.2	70.5	73	7.209			.785
.448	-1.9	+1.7	50.2	70.4	74	7.200			.783
.509	0.0	0.0	50.2	70.5	74	7.162			.793
.577	+2.3	-2.4	50.2	70.7	74	7.044			.818
						7.060			
.650	+4.6	-4.8	50.2	69.8	71	6.742			.807

ave
(.794)

RUN #51
25 Nov 70
P_T = 14.6 psig +1

P_{CH} = 2.1 psig
P_W = 120 psig
P_S = 40 psig

T_{CH} = 218°
T_W = 179°
V = 25.8

.30 orifice
46 GPM recirculation
2 mm cooling
3 mm overplate

12/P _T	P _T	L	R	T _{IN}	T _{OUT}	T _{CH}	T _{CH}	T _{CH}	WT	WT	X
.119	14.6	-12.2	+12.6	50.2	71.2	76	7.317	105.1	323.1	982.4	.818
								218.0			
.192	14.8	-9.8	+10.2	50.1	70.6	76	7.314				
							7.306			961.5	.786
.261	15.0	-7.5	+7.9	50.2	70.8	76	7.304			966.2	.791
.321	15.1	-5.6	+5.8	50.2	70.8	76	7.300			966.7	.791
.374	15.3	-3.6	+3.8	50.2	70.9	75	7.288			972.7	.798
.434	15.4	-1.8	+1.9	50.2	70.7	76	7.298			962.5	.786
.500	15.5	+0.4	-0.4	50.1	70.8	76	7.295			971.8	.796
.555	15.7	+2.3	-2.4	50.2	70.7	76	7.268			966.3	.790
.621	16.1	+4.7	-5.0	50.2	70.7	76	7.178			977.9	.802
.645	16.8	+7.7	-8.1	50.2	70.3	75	6.964			986.8	.811
.769	18.1	+11.4	-11.7	50.0	69.5	73	6.674			996.4	.820

ave
.799

Run # 52

27 Nov 70

 $P_T = 14.6 \text{ psig}$ $P_{\text{cell}} = 1.5 \text{ psig}$ $P_W = 150$ $P_2 = 79$ $T_{\text{cell}} = 218 \text{ } ^\circ\text{F}$ $T_W = 188 \text{ } ^\circ\text{F}$ $V = 25.7 \text{ mL}$ $.30 \text{ g/cell}$ 60.6 g/H NO₂ w/PLP2 min cooling
3 min sample

P/P _T	L	R	TIN	TOUT	TCELL	CEN	TIRE	WT	X
.172	-12.1	+12.5	50.1	71.7	76	7.515	105%	312 ³	.769
						7.493		207.7	
.222	-8.9	+9.3	50.0	71.1	76	7.587			
						7.429			.752
.333	-5.4	+5.6	49.9	71.5	77	7.385			.757
.383	-3.8	+4.0	49.9	71.2	75	7.369			.768
.445	-7.7	+1.9	49.9	70.9	75	7.374			.753
.499	-0.1	+0.2	49.6	70.9	76	7.325			.774
.574	+2.2	-2.3	49.6	70.7	75	7.196			.781
.658	+4.8	-5.1	49.6	69.9	74	6.959			.775

(ave)
.765

RUN #53
15 DEC 70
P = 14.6 psig

PCH = 3.1 psig T_{CH} = 220°F
P_W = 132 psig T_W = 177°F
P_b = 118 psig V = 25.5 in

.30 c/ice
800 GPM CCS
2 min cooling
2 min sample

P/p	L	R	T _{IN}	T _{OUT}	T _{CH}	CON	THICK	WT	X
.115	-12.4	+12.7	47.1	66.7	71	7.700	104 ²	332 ²	.540
						9.641	227.5		.545
.177	-10.4	-10.7	47.1	66.6	71	9.897			.521
						9.927			
.264	-7.6	+7.9	47.0	66.6	71	9.732			.538
.322	-5.7	+6.0	47.1	66.7	70	9.682			.540
.400	-3.3	+3.4	47.1	66.3	70	9.691			.525
.448	-1.8	+1.8	47.1	66.3	70	9.671			.526
.509	0.0	0.0	47.1	66.1	70	9.640			.521
.568	+2.0	-2.1	46.9	65.9	69	9.512			.530
.650	-4.6	-4.8	46.9	65.5	68	9.365			.526

ave
(.531)

LUN #55

17 Dec 70

$$P_T = 5.3 \text{ g/sing}$$

$P_{4E} = 3.0 \text{ psig}$ $T_{4E} = 221^\circ \text{F}$

$T_w = 139^\circ \text{F}$ $T_w = 178^\circ \text{F}$

$\rho_s = 117 \text{ g/cc}$ $V = 26.2 \text{ in. Hg}$

.30 office

60 GPH 12000

2 mm coating

Forster & Co

θ^3/P_T	L	R	TIN	TOUT	T-ON	CON	TRIP	WT	X
.143	-12.9	+13.3	46.4	66.3	60	9.671	109'	270°	.357
							(165.8)		
.204	-11.6	+11.9	46.4	66.4	61	9.781			
						9.761			.357
.292	-9.6	+10.0	47.9	67.1	61	9.710			.338
.381	-7.7	+8.0	48.0	66.9	61	9.695			.333
.451	-6.1	+6.5	48.2	67.0	61	9.627			.332
.525	-4.5	+4.8	48.3	67.1	62	9.567			.336
.589	-3.2	+3.3	48.4	66.8	62	9.476			.330
.646	-1.9	+2.1	HOSE ROCK						Ave
									.390

.30 a/jice
60 GVA 100%
2 min cooling
2 min sample

$\frac{P}{T}$	L	R	T.N	Tout	T.W	C.W	Time	WT	X
.120	-13.4	+13.8	48.3	68.7	63	7.871	105 ^g	261 ³	.462 (156.5)
.192	-11.8	+12.2	48.3	68.8	63	8.059 7.785			.472
.267	-10.2	+10.5	48.5	69.0	63	7.836			.468
.342	-8.5	+8.9	48.5	69.0	63	7.766			.473
.399	-7.3	+7.6	48.5	68.9	63	7.792			.467
.490	-5.3	+5.6	48.5	68.7	63	7.803			.461
.533	-4.4	+4.6	48.5	68.6	63	7.753			.462
.621	-2.4	+2.7	48.5	68.2	63	7.563			.465
.732	-0.1	+0.1	48.6	66.7	62	7.227			.439

(ave)
.463

RUN #58
21 DEC 70
P=5.3 psig

PAL = 30 psig
PN = 156 psig
PS = 112 psig

TCAL = 219°F
TW = 197°F
V = 26.0 in Hg

.30 cr/ft
4064 H₂O/gal w/PLMS
2 min Cooling
3 min Sample

P_{it} L R T_{IN} T_{CUT} T_{cen} C_{en} T_{AE} WT X

.124	-13.3	+13.7	46.5	68.0	61	6.465	103 ² 259 ³	.637
							156.2	
.206	-11.5	+11.9	46.4	67.4	61	6.656		.596
.279	-10.0	+10.4	46.4	67.5	61	6.981		.670
.351	-8.3	+8.7	46.5	67.8	61	6.410		.637
.424	-6.7	+7.1	46.5	67.4	61	6.621		.596
.487	-5.3	+5.7	46.5	67.0	60	6.913		.605
.537	-4.3	+4.5	46.5	66.8	60	6.287		.613
.639	-2.1	+2.2	46.5	66.7	60	5.811		.673
.707	+0.6	-0.7	46.6	65.2	59			

(ave)
.622

RUN #59
22 DEC 70
R = 5.3 psig

P_{CH} = 3.1 psig
P_W = 137 psig
P_S = 117 psig

T_{CAV} = 222°F
T_W = 197°F
V = 270 in Hg

.30 or less
17.5 GPH/RLP
2 min cooling
3 min sample

P/R	L	R	T _{IN}	T _{OUT}	T _{CAV}	CON	TAPE	WT	X
.124	-13.3	+13.7	46.2	64.3	59	5.048	1043	2729	.874
.147	-11.7	+12.1	46.2	63.7	58	5.377		187.5	.776
.297	-9.5	+9.9	46.2	63.9	59	4.979			.865
.397	-8.9	+8.8	46.3	64.1	58	4.761			.890
.415	-6.9	+7.3	46.4	63.9	58	5.265			.796
.471	-5.7	+6.0	46.4	63.6	58	5.251			.782
.547	-3.6	+3.8	46.5	63.7	58	4.505			.941
.639	-2.1	+2.2	46.5	63.6	58	4.600			.912

(ave
.854)

RUN #60
 22 DEC 70
 R: S3 P315

RA = 3.0355 T₄ = 281°F

.30 of ice
 Dry Run
 2 min cooling
 3 min sample

P/P _T	L	R	T _{IN}	T _{OUT}	T _{CON}	T _{PRE}	WT	X
.106	-13.7	+14.1	45.1	63.4	56	4.079	105.2	271.3
							186.8	
.183	-12.0	+12.9	45.0	62.6	56	4.503		.961
.287	-9.7	+10.1	44.9	62.7	56	4.071		1.017
.390	-7.5	+7.8	44.8	63.1	56	4.159		1.105
.481	-5.5	+5.8	44.8	62.4	56	4.057		1.086
.557	-3.8	+4.1	44.8	62.5	55	3.979		1.118
.684	-1.1	+1.2	44.8	61.6	55	3.906		1.075

ave
 (1.082)

Thesis
H89

Hulsizer

125781

Critical flow of
wet steam through a
well rounded orifice.

22 APR 71

DISPLAY

Thesis
H89

Hulsizer

125781

Critical flow of
wet steam through a
well rounded orifice.

thesH89missing

Critical flow of wet steam through a wel



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C.I.

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